ADVANCED EXERGY AND EXERGOECONOMIC ANALYSIS OF THE MAJOR COMPONENTS OF A COMBINED CYCLE POWER PLANT

A Thesis

by

COREY J. BROWN

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Chair of Committee,	Jerald A. Caton
Committee Members,	Charles H. Culp
	Bryan P. Rasmussen
Head of Department,	Andreas A. Polycarpou

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ABSTRACT

For a power plant, or any energy conversion system, exergetic analysis methods are used to determine the irreversibilities, or exergy destruction, in the system components. Advanced exergetic analysis methods indicate that the irreversibilities within a component can be considered in two parts. The first part, which is due to irreversibilities within the component, is referred to as the endogenous exergy destruction. The second part, which is due to component interactions, is referred to as the exogenous exergy destruction. While some of the irreversibilities within an energy conversion system can be avoided by improving the performance of individual components or the system structure, due to technological and economic constraints, some are unavoidable. Newer advanced exergetic analysis methods seek to determine the realistically avoidable exergy destruction for each component in the system. Exergoeconomic methods seek to determine the cost-effectiveness of improving certain components. In this thesis, conventional and advanced methods of energy, exergy, and exergoeconomic analysis are used to evaluate the performance of the major components of a combined cycle power plant.

The conventional and advanced exergetic analyses reveal that the combustion chamber is the largest source of exergy destruction within the system, as well as the largest source of endogenous exergy destruction that can be avoided. The exergoeconomic analysis reveals that the combustion chamber is also the most cost-effective component to improve, due to its relatively low cost of capital investment and relatively high cost of exergy destruction. By reducing the air-fuel ratio, and further preheating the combustion air, 29.25 MW of exergy destruction in the combustion chamber can be avoided. This would also lead to increased performance in the expander by increasing its inlet temperature. The advanced exergy analysis also reveals that the expander is the second largest source of avoidable endogenous exergy destruction. However, the exergoeconomic analysis reveals that from a cost basis there is little room for improvement of the component. The results obtained provide useful information for the optimization of the power plant in question.

DEDICATION

This thesis is dedicated to my mother and late grandfather, without whom none of my success would have been possible.

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I would like to express my sincere appreciation to my thesis advisor Dr. Jerald Caton for his support and guidance throughout my research, and graduate education. He has gone above and beyond to help me complete my education at Texas A&M, and I could not have asked for a better thesis advisor.

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NOMENCLATURE

Symbols

c	Cost rate per unit exergy ($/MMBtu)
C	Economic constant
$\dot{C}_D$	Cost rate of exergy destruction ($/h)
e	Specific exergy (Btu/lb)
$\dot{E}$	Rate of energy (MW)
$\dot{E}_D$	Rate of exergy destruction (MW)
$\dot{E}_F$	Rate of fuel exergy (MW)
$\dot{E}_L$	Rate of exergy loss (MW)
$\dot{E}_P$	Rate of product exergy (MW)
f	Exergoeconomic factor (%)
°F	Degrees Fahrenheit
h	Specific enthalpy (Btu/lb)
I	Investment cost (USD)
K	Kelvin
$\dot{m}$	Mass flow rate (klb/h)
P	Pressure (psia)
R^2	Coefficient of determination
t	Time (seconds)
T	Temperature (°F)

$\dot{W}$	Work (MW)
x_D	Exergy destruction fraction
y	Exergy destruction ratio (%)
$\dot{Z}$	Levelized investment cost ($/h)
ε	Exergetic efficiency (%)
η	Isentropic efficiency (%)

Abbreviations

AC	Air compressor
CC	Combustion Chamber
CT	Combustion turbine
CCPP	Combined cycle power plant
CCPP1	Combined cycle power plant #1
CCPP2	Combined cycle power plant #2
CGAM	Acronym for names of researchers
DC2	DC2 air cooler
EXCEM	Exergy, cost, energy, and mass analysis
GT	Expander
HPST	High pressure steam turbine
HRSG	Heat recovery steam generator
IGV	Inlet guide vanes
IPST	Intermediate pressure steam turbine

LPST	Low pressure steam turbine
ST	Steam turbine
SPECO	Specific exergy cost analysis
TIT	Turbine inlet temperature
TOT	Turbine outlet temperature
USD	US dollars

Subscripts/Superscripts

a	Air
ad	Adiabatic
AV	Avoidable
ch	Chemical
CI	Cost of Investment
cv	Control volume
e	Exit
D	Destruction
EN	Endogenous
EN,AV	Avoidable endogenous
EN,UN	Unavoidable endogenous
EX	Exogenous
EX,AV	Avoidable exogenous
EX,UN	Unavoidable exogenous
F	Fuel

i	Inlet
in	Inlet
j	j-th component
k	k-th component
loss	Loss
min	Minimum
OM	Operations and maintenance
out	Outlet
P	Product
ph	Physical
q	Heat transfer
s	Isentropic
tot	Total
w	Work
0	Dead state

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1. INTRODUCTION

1.1: Need for Improving Power Plant Performance

Power plants, much like any other business, are first and foremost driven by economics. In the US, a city of 3000 homes requires roughly 300 MW of electricity per day [1]. At a fuel cost of $3USD/MMBtu, it would cost a combined cycle power plant about $55 million USD per year to produce this much power. Reducing the power plant's fuel usage by just 2%, would save the plant over a million dollars annually. The environment is also of concern, as power plants produce carbon dioxide, nitrogen oxides and other harmful pollutants. Burning coal and certain fossil fuels releases a higher concentration of pollutants into the atmosphere than alternative fuels such as natural gas and renewables. In an effort to combat this issue, energy producers are slowly moving away from coal towards natural gas and renewables. Natural gas is projected to be used in 37% of the electrical power generation in the US by 2035, surpassing coal as the most widely used fuel for this application [1]. For these and other reasons, the continued advancement and optimization of power plants, especially those which run on alternative fuels, is of the utmost importance.

Power plant performance can be improved by improving the overall system design, the individual component performance, and the management of the plant operation. In order to find the areas with the most potential for improvement, a comprehensive analysis should be conducted. Improving the methods of analysis can translate to further optimization of the systems. Traditionally, these analyses have been largely based on the first law of thermodynamics. More recently, power plant performance has been examined from the perspective of the second law of thermodynamics. A second law based analysis is introduced in the following section.

1.2: Exergy Analysis

The second law of thermodynamics has several definitions, but in essence it declares that energy has quality, and that actual thermodynamic processes occur in the direction of decreasing quality. Exergy is based on the second law and is a measure of a system's maximum work potential with respect to the surrounding environment. Exergy destruction is a measure of the work potential lost due to irreversibilities in the system. Various exergy, or exergetic, analysis methods provide engineers a way to determine the location, magnitude, and sources of irreversibilities. Engineers have continuously worked to improve these methods, especially in the past decade. A new and useful exergy analysis method is that of splitting the exergy destruction of the system components into its avoidable and unavoidable components, and endogenous and exogenous components. Endogenous exergy destruction is due to irreversibilities within the component, whereas exogenous exergy destruction is due to interactions with other components of the system. By splitting the exergy destruction in this fashion, the sources of realistically avoidable irreversibilities can be pinpointed. With this information, plant layout can be improved, capital and operational costs can be decreased, and more efficient operating schedules can be devised.

1.3: Exergoeconomic Analysis

Economics are what truly drive the continued optimization of power conversion systems, and thermoeconomic, or exergoeconomic, methods combine economics with the exergy analysis. By providing the estimated costs (USD) due to exergy destruction, exergoeconomic analysis allows for the improvement of plant design and operation from an economic standpoint. The method presented in [2, 3] provides additional information by splitting the cost of exergy destruction into its endogenous and exogenous components, then further into it avoidable endogenous and exogenous components, and unavoidable endogenous and exogenous components. From the avoidable and unavoidable endogenous and exogenous exergy destruction of each component, the most cost-effective way to improve can be found.

1.4: Objectives

In the following thesis, conventional energy based methods are first used to determine the variation in performance between two similar natural gas combined cycle power plants (CCPP). Operational data with measured gas turbine compressor inlet air mass flow rate were used; a parameter not available in previous studies of this nature. One of the plants is then further evaluated using conventional and advanced exergy and exergoeconomic methods, and the plant components with the most potential for improvement are discussed. The advanced exergy and exergoeconomic methods [2, 3] that are used were developed within the past decade, so there is little replication in the literature.

2. LITERATURE REVIEW

The following literature review focuses on the exergy, exergoeconomic, and sensitivity analyses. A representative collection of works and their related findings are presented, with greater emphasis on papers from the past decade.

2.1: Conventional Exergy Analysis

A basic definition of exergy, also referred to as work potential and available work, is the potential of a substance to do work. While in thermodynamic equilibrium with its surroundings, a substance has no potential to do work, and therefore its exergy is zero. Sciubba and Wall [4] defined exergy as the maximum theoretical useful work obtained if a system S is brought into thermodynamic equilibrium with the environment by means of processes in which S interacts only with this environment. Unlike an energy based analysis, an exergy analysis allows one to identify the location, magnitude, and origin of thermodynamic inefficiencies. This additional information is useful for improving the efficiency and cost-effectiveness of a system, and providing means of comparing multiple systems.

The idea of available work can be traced back almost two hundred years. According to Sciubba and Wall [4], Carnot (1824) first stated that the work that can be extracted from a heat engine is proportional to the temperature difference between the hot and the cold reservoir. However, Clapeyron (1832, 1834), Rankine (1851), Thomson (1852), and Clausius (1850, 1867) all contributed to the establishment of the principle.

The earliest mention of the term available work was by Gibbs [5] in 1873. Gibbs modeled an arbitrary substance as a triangular surface. He said that the surface is made up of three points, the properties of the substance: volume, energy, and entropy. The surface is acted upon by the two other properties, temperature and pressure. When the substance is in thermodynamic equilibrium with its surroundings, the triangular surface is tangent to the plane; if it is not, then it is shifted, and its coordinates change. The movement is parallel to the respective axis of each property. The coordinates for which the surface is tangent to the plane, and the substance is in thermodynamic equilibrium with its surroundings, will be referred to as at the dead state. Exergy is basically a measure of the distance between the coordinates of the old and new vertices of the triangular surface. Therefore the work potential equals the sum of these three distances, with the sign of the entropy term being negative.

It is the job of the power plant to extract as much of the inlet substances' work potential, and convert it to useable energy. Giannantoni et al. [6] proposed the upgrade of a steam power plant to CCPP using a five part approach: energy analysis, conventional exergy analysis, thermoeconomic analysis, environmental evaluation, and economic evaluation. They found the energy analysis to be the most helpful, however that is likely no longer the case, as exergy and thermoeconomic analysis methods have improved in the years since.

Butcher and Reddy [7] performed a conventional energy and exergy analysis of a waste heat recovery power generation system. They found that the first and second law efficiencies of the HRSG decrease with increasing pinch point, and that higher gas inlet temperatures lead to an increase in second law efficiency of the HRSG.

Çolpan [8] applied a conventional exergy and thermoeconomic analysis to a combined-cycle cogeneration plant, using the SPECO method and CGAM problem. De Sa and Al Zubaidy [9] investigated the effect of ambient temperatures on the energetic performance of a gas turbine. They found that for every degree Kelvin rise in compressor inlet temperature, there was a 0.07% drop in combustion turbine (CT) thermal efficiency.

Kaushik et al. [10] conducted an in-depth review of the energy and conventional exergy analysis as applied to coal and gas power plants, noting the method of splitting the exergy destruction into endogenous and exogenous components as a potential improvement.

Song et al. [11] studied the exergetic performance of a CT and had several interesting results. They found the largest exergy destruction was due to chemical reactions in the combustor. The variable inlet guide vanes (IGV), despite being used to enhance part load performance, caused a significant increase in exergy destruction at the first compressor row, and a decrease in overall compressor efficiency. Also, turbine blade cooling air accounted for over half the total exergy destruction in cooled stages, significantly affecting overall turbine efficiency.

Acir et al. [12] looked at the effect of varying dead state temperatures on energetic and exergetic efficiencies in a thermal power plant. They found the exergy efficiency to decrease by ~8% for an ~8% increase in dead state temperature (278K to 303K).

Aljundi [13] investigated the effects of varying the dead state conditions at a steam power plant in Jordan. He found the majority of irreversibilities to be in the condenser, and little change in performance of the main components due to varying dead state temperature.

Gülen and Smith [14] derived a simple method for estimating the Rankine bottoming cycle output directly from the CT exhaust exergy. In a separate paper, Gülen and Joseph [15] said that a combined cycle plant will run under boundary conditions significantly different from those under which individual components are designed. They also presented a simple, generalized, physics-based calculation method to estimate off-design performance.

Aklilu and Gilani [16] performed a conventional exergy analysis on a cogeneration plant running at varying part loads. They found the exergy destruction rate of the CC to be 50.6% to 63.7% of the overall system destruction rate, making it the largest source of destruction. They attributed the majority of destruction the chemical reaction and mixing of dissimilar fluid streams.

Haseli et al. [17] performed a conventional exergy analysis of a solid oxide fuel cell combined with a recouperative gas turbine plant. They found that increasing the turbine inlet temperature (TIT) leads to an increase in power output, however, it also leads to a reduction in the overall cycle's thermal and exergy efficiency. An increase in either TIT or compression ratio resulted in an increase in cycle exergy destruction.

2.2: Advanced Exergy Analysis

In her dissertation, Kelly [2] presented a complete method and application of an advanced exergy analysis which splits the exergy destruction into its exogenous and endogenous parts, and its avoidable and unavoidable parts. Using this method, it was found that improving the CT air preheater is more profitable than improving CT compressor, contrary to the conventional exergoeconomic analysis. Suggestions including reducing pressure drop, increasing heat transfer in the air preheater, and preheating the fuel, were made to improve the component with the most potential savings, the combustion chamber.

Kelly et al. [3] further described the method for splitting exergy destruction into exogenous and endogenous components, which illustrate the component irreversibilities, as well as the irreversibilities of a component due to the entire system design. These parameters were be split a second time to illustrate the avoidable and unavoidable irreversibilities. The unavoidable irreversibilities are due to technological limitations such as availability and cost of materials and components, however, this value is somewhat arbitrary as the limitations must be estimated. Nevertheless, the advanced exergy analysis provides deeper insight into system performance and component interactions.

Petrakopoulou et al. [18] used conventional and advanced exergy methods to conduct an environmental evaluation of a power plant. They found that as result of being the largest source of exergy destruction, the combustion chamber has the most environmental impact, and 68% of the exergy destruction is unavoidable. In [19], Petrakopoulou et al. applied conventional and advanced exergy methods to a combined cycle power plant and found the combustion chamber to be the largest source of irreversibility in the system. They determined the combustion chamber should be improved first, followed by the expander and air compressor.

In [20], Wang et al. applied conventional and advanced exergy methods to a supercritical power plant. From their results of splitting the exergy destruction, they suggested the generator and steam turbines should be the first components to be improved.

In [21], Morosuk and Tsatsaronis outlined the strengths and limitations of the advanced exergy analysis. They presented three important questions that should be answered in order to thermodynamically improve an energy conversion system:

1. What is the maximum possible decrease of the exergy destruction within each system component?
2. How will reducing the exergy destruction of the k-th component affect the exergy destruction of the other components in the system?
3. Are there any other ways to restructure the system so that the exergy destruction of the k-th component, or more importantly, within the overall system can be reduced?

These questions are not addressed in a conventional exergy analysis, however, they are addressed in certain advanced exergy analyses that split the exergy destruction term. The limitations of advanced exergy analysis include arbitrariness of some values, a need to simulate unique processes, and the requirement of many simulations.

2.3: Exergy Economics

According to Sciubba [4], the earliest idea of combining thermodynamics and economics was by Lotka in 1921, and the application of exergy analysis and engineering economics was proposed in the early 1960's. In Europe it was called exergo-economics (Rabek 1964, Szargut & Petela 1964, Baehr et al. 1965, Brodyanski 1965, Fratzscher 1965, Elsner 1965, Nitsch 1965, Bergmann & Schmidt 1967) and in the US it was called thermos-economics (Evans 1961, Tribus 1961, Tribus & Evans 1962, Evans & Tribus 1965, El-Sayed 1970).

With respect to the modern exergoeconomic analysis, Tsatsaronis, Lazzaretto, Diner, and Rosen are some of the main researchers. In 1984, Tsatsaronis [22] first defined the term exergoeconomics as a more specific term for an exergy based thermoeconomic analysis. In 1985, Tsatsaronis and Winhold [23, 24] described a new method of exergy based thermoeconomic analysis, or exergoeconomic analysis, for power plants. This method is broken down into seven steps:

1. Conduct detailed mass, energy, and exergy balances of the plant.
2. Calculate the investment and operating costs for each plant component.
3. Calculate the cost of the exergy unit of each process flow stream.
4. Calculate the average exergy unit cost of fuel and products of each component.
5. Calculate the cost of the exergy losses in each component.
6. Interpret the results.
7. Conduct a sensitivity analysis and make recommendations.

This method of exergoeconomic analysis is useful for illustrating the sources of costs, providing a means of comparison between them, and calculating the optimum capital cost to exergy loss ratio for a given design. This method can be used to improve decisions concerning selection and optimization of process design, plant maintenance, and replacement of certain plant components.

Valero et al. [25] presented the application of four methods of thermoeconomic analysis and optimization, called the CGAM problem, an acronym for the principle researchers' first names. The CGAM problem consists of a physical model, thermodynamic model, and economic model, applied to a small cogeneration plant. Decision variables are chosen to be the pressure ratio, the air compressor efficiency, the gas turbine efficiency, the air temperature at the preheater exit, and the combustion gas temperatures at the turbine inlet. The aim of the CGAM problem is to unify the four thermoeconomic methodologies as each has specific fields of applications for which it provides proven and efficient solutions.

In their text, Dincer and Rosen [26] presented the necessary equations for conducting a conventional exergy analysis of several systems, including combined cycle. They also explained two methods of exergoeconomic analysis: exergy, cost, energy, and mass analysis (EXCEM) and specific exergy cost analysis (SPECO). Lazzaretto and Tsatsaronis [27] presented a final form of the SPECO method to define exergetic efficiencies, and calculate the auxiliary costing equations for thermal system components.

Cziesla et al. [28] determined the avoidable thermodynamic inefficiencies and costs in an externally fired combined cycle power plant. They found that the largest avoidable cost of exergy destruction occurs in the combustion chamber, but that the component had a low potential for improvement.

Ahmadi et al. [29] performed a comprehensive conventional exergy, exergoeconomic, and environmental impact analysis comparing several combined cycle power plants (CCPP). They found the combustion chamber to have the most significant exergy destruction and cost, and the amount of supplementary firing to be proportional to the CCPP exergy efficiency.

Karaali and Oztürk [30] introduced a simple method of thermoeconomic optimization and applied it to four cogeneration cycles with constant power output. They found that there is an optimum excess air flow rate for each cycle that gives the minimum electricity cost.

Kelly [2] used the advanced exergy method to improve upon existing exergoeconomic analysis methods. Most notably, by calculating cost rates from the avoidable endogenous and exogenous exergy destruction rates of the major component, a realistically attainable potential savings can be found, as well as the source of the destruction. By having the avoidable costs associated with each component, and knowing whether they are due to the plant configuration or inefficiencies in the component itself, a more informed decision can be made with regard to system improvements.

Clearly power plants have been thoroughly studied using conventional methods. However, because they are newer, and require numerous simulations, few researchers have applied the advanced methods of Kelly et al. [2, 3] in their analyses.

3. BACKGROUND

3.1: Combined Cycle Power Plant

In 1882 Thomas Edison implemented the first central power plant, Pearl Street Station; a cogeneration plant [31]. In the years since, power generation has played a key role in everyday life. With rising energy costs, increased plant efficiency has become increasingly important and desirable. Even with the most efficient components, simple cycle power plants are still wasting about half of their energy. For many years, engineers have researched a multitude of ways to reduce wasted energy, and harness the most work potential possible. Of these methods, cogeneration and combined cycle power plants have emerged as two of the top contenders. Combined cycle power plants utilize waste heat from a prime mover to produce steam to power a steam turbine for further generation of electricity. In the case of CCPP1 and CCPP2, the prime mover was a combustion gas turbine.

3.2: Combustion Turbine

According to Boyce [32], technological advances have spurred the increased use of combustion gas turbines, or combustion turbines (CT), in power and petrochemical industries in the past thirty years. Since the introduction of the combustion turbine, thermal efficiency has increased from 15% to over 45%. With new combustion turbines exceeding thermal efficiencies of 45%, combined cycle plants can reach overall efficiencies of 60%. These higher turbine efficiencies are possible because of new schemes of air-cooling and improvements in blade materials; allowing for higher temperatures and pressure ratios.

The combustion turbine (CT) consists of three major components, the air compressor (AC), the combustion chamber (CC), and the expander (GT). With regard to the air compressor, two major factors dictate CT power output: mass flow rate and compressor efficiency.

When the turbine is running at part load, inlet guide vanes (IGV) regulate the air intake. According to Davis [33], at a fixed IGV angle, and synchronous speed, the gas turbine compressor is well approximated as having a constant volumetric flow rate. At constant volumetric flow rate, mass flow rate depends on inlet air density, which is most affected by air temperature. Some common ways to marginalize the negative impact of a higher inlet temperature are evaporative cooling, fogging, and chilling of the compressor inlet air. Reducing the inlet air temperature results in increased air density, thereby increasing mass flow rate and power output. However, relative humidity also affects air density, with an inverse relation. Therefore, the ideal operating point is when the compressor inlet temperature is at a minimum, while maintaining a low enough relative humidity. Chillers do not directly inject water into the air, making them the most effective in that regard.

An increase in the pressure differential across the compressor inlet air filter results in a reduction in mass flow rate and power output. Therefore, clean filters are crucial for optimal system performance. Extractions also result in reduced mass flow rate, and a trade-off must be made between the amount of turbine blade cooling, and mass flow rate through the compressor. As they are metal, if the turbine blades get too hot, they could expand, and cause catastrophic failure.

A major feature that separates the following analysis from those previously conducted was the implementation of measured compressor inlet air mass flow rate data, accurate within 1%. Due to non-disclosure agreements, the method by which this is achieved cannot be disclosed.

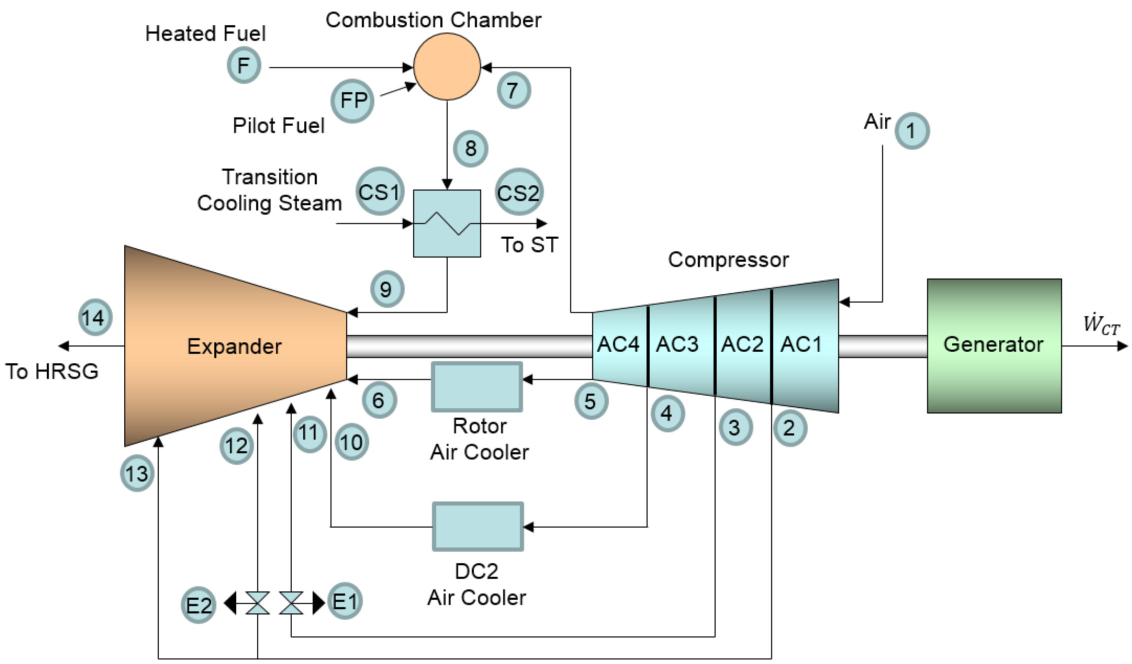

Figure 3.1: Siemens/Westinghouse W501G

CCPP1 and CCPP2 are both natural gas powered combined cycle power plants in humid subtropical climates. For CCPP1, the total possible electrical power generated is 746 MW from two gas turbines and one steam turbine. For CCPP2, the total possible electrical power generated is 343 MW from one gas turbine and one steam turbine.

CCPP1 and CCPP2 both use Siemens/Westinghouse W501G industrial gas turbines. A diagram of this machine presented in figure 3.1. The W501G compressor houses 16 rows of blades, with four possible extractions for gas turbine blade cooling. The first low pressure extraction occurs at row six. This extraction can be used to cool the fourth row of gas turbine blades, cool the gas turbine exhaust gas temperature, or bled off. The second extraction occurs at row eleven, and can be used to cool gas turbine row three or bled off. The third extraction occurs at high-pressure, high-temperature row thirteen, and is sent to an air cooler (DC2). Air exiting the DC2 air cooler is used to cool turbine row two. The final extraction occurs at high-pressure, compressor row sixteen. The air is routed through a rotor air cooler to the gas turbine inlet.

The remaining air from the compressor is sent to the combustors to be heated. After combustion, the product gasses are sent through a heat exchanger, where steam slightly cools the mixture. The high pressure exhaust then enters the turbine, mixing with air from the rotor air cooler, to produce mechanical power. A fraction of this energy is used to power the compressor, and the rest to generate electricity via the generator.

For the CT, there are a few major factors that most impact overall performance. These factors include AC inlet air mass flow rate, AC inlet air temperature, AC pressure ratio, GT inlet temperature (TIT), and GT pressure ratio. Therefore, in the 1st law analysis to follow, focus will be put on these parameters' effect on CT performance, namely power output and isentropic efficiency.

The cleanliness of the AC inlet air filter can have an effect on the performance of the AC, which can be quantified by measuring the differential pressure across the filter.

The difference between the ambient pressure, and static pressure in the AC inlet duct, is referred to as the AC inlet differential pressure.

The TIT has a significant effect on GT performance. At a lower temperature, the combustion exhaust gases entering the turbine have less enthalpy and work potential. If the TIT is reduced, and the turbine outlet temperature (TOT) is held constant, the work output of the GT will decrease. If the TIT is reduced, and the work output is held constant, the TOT will decrease. The latter case results in the GT exhaust gas having work potential when it enters the HRSG, reducing any power producing downstream of the GT. A change in TIT also affects the combustion chamber, and potentially components farther upstream.

3.3: Heat Recovery Steam Generator

The heat recovery steam generator (HRSG) is a vital piece of equipment in combined cycle and CHP power plants. In the case of CCPP1 and CCPP2, the HRSG uses the gas turbine exhaust to produce high temperature, high-pressure steam for use in the steam turbine. The HRSG is divided into three sections. Each section contains a preheater, an evaporator, and a one or two stage super heater. Superheating the steam before it enters the steam turbine leads to greater efficiency and energy production. There are three main types of HRSGs: unfired, supplementary fired, and fully fired. Unfired models use only the exhaust energy to produce steam. Supplementary and fully fired models use additional fuel to increase the production of steam. CCPP1 and CCPP2 are supplementary fired.

In HRSG design, many factors must be considered. According to Boyce [32], the pinch point is the difference between the exhaust temperature leaving the evaporator section and the saturation temperature of the steam**.** A lower pinch point will lead to the recovery of more heat. However, lower pinch points require more surface area for heat transfer to occur, increasing cost and back pressure. Also, very low pinch points can lead to inadequate steam production. An effective pinch point is usually between 40°F and 60°F. Approach temperature is the difference between the saturation temperature of the steam, and the inlet water. A lower approach temperature can increase steam production, but will increase cost. These are typically between 20°F and 50°F. Economic analysis must be done to evaluate the optimum combination of surface area, gas pressure drop, and steam production.

3.4: Steam Turbine

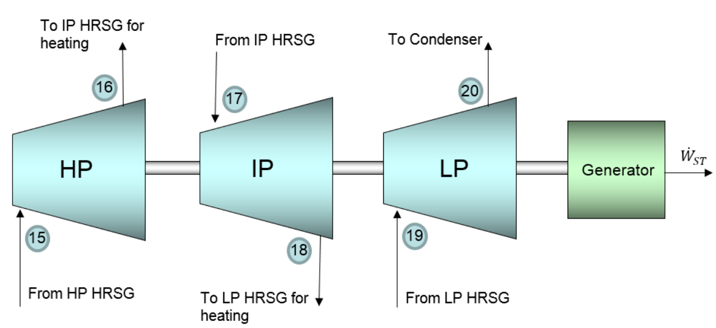

Figure 3.2: Three-pressure steam turbine

CCPP1 and CCPP2 both use three stage, condensing steam turbines with reheat. High pressure, superheated steam leaves the heat recovery steam generator and enters the high pressure steam turbine inlet, where it is expanded to an intermediate pressure.

The intermediate pressure steam is then reheated in the HRSG, and injected with a fraction of the transition cooling steam from the gas turbine cycle. The steam is then expanded through the intermediate pressure turbine to a low pressure. This low pressure steam is once again reheated in the HRSG, and injected with the remaining transition cooling steam. Lastly, the steam is expanded through the low pressure turbine and partially condenses. The system is controlled to maintain a high enough steam quality so that deterioration and fouling of the low pressure turbine blades is kept to a minimum.

4. METHODOLOGY

In this section, the basic methodology for the conventional and advanced energy, exergy, and exergoeconomic analysis is presented for the major components and overall power plants. As CCPP1 and CCPP2 use similar components, the same methodology is used for the analysis of both power plants' components. At the system level, the additional gas turbine at CCPP1 simply contributes additional components to the overall system equations.

4.1: Energy Analysis

Energy analysis based on the First Law of Thermodynamics is the most commonly used method for power plant performance diagnostics. Using the methodology described in this section, efficiencies and work interactions were calculated for all of the compressors, expanders, and combustors.

The first step in the energy analysis of the CCPP1 and CCPP2 was the mass flow balance, the general equation, from [34], is presented in equation 4.1 for any component k. The mass flow balance is presented in equations 4.2 through 4.5 for the AC, CC, GT, and LPST, respectively. While equations are only presented for the LPST, the same methodology applies to the IPST and LPST stages.

$$\sum \dot{m}_{in,k} = \sum \dot{m}_{out,k} \tag{4.1}$$

$$\dot{m}_1 = \dot{m}_7 + \dot{m}_2 + \dot{m}_3 + \dot{m}_4 + \dot{m}_5 \tag{4.2}$$

$$\dot{m}_7 + \dot{m}_F + \dot{m}_{FP} = \dot{m}_8 \tag{4.3}$$

$$\dot{m}_6 + \dot{m}_9 + \dot{m}_{10} + \dot{m}_{11} + \dot{m}_{12} + \dot{m}_{13} = \dot{m}_{14} \tag{4.4}$$

$$\dot{m}_{19} = \dot{m}_{20} \tag{4.5}$$

The next step in the analysis was an energy balance, presented in equation 4.6 for any component k. By applying the assumptions listed in table 4.1, equation 4.6 was simplified to equations 4.7 through 4.10 for the AC, CC, GT, and LPST, respectively.

Component	Assumptions
Air Compressor (AC)	-adiabatic, steady flow, and negligible potential and kinetic energy effects
Combustion Chamber (CC)	-adiabatic, steady flow, ΔP=0, complete combustion
Expander (GT)	-adiabatic, steady state, steady flow, negligible potential and kinetic energy effects
High-pressure Steam Turbine (HPST) Intermediate-pressure Steam Turbine (IPST) Low-pressure Steam Turbine (LPST)	-adiabatic, steady state, steady flow, and negligible potential and kinetic energy effects

Table 4.1: Assumptions for the 1st Law analysis

$$\sum \dot{E}_{in,k} - \sum \dot{E}_{out,k} = \frac{\Delta \dot{E}_k}{\Delta t} \tag{4.6}$$

$$\dot{W}_{AC} + \dot{m}_1 h_1 = \dot{m}_2 h_2 + \dot{m}_3 h_3 + \dot{m}_4 h_4 + \dot{m}_5 h_5 + \dot{m}_7 h_7 \tag{4.7}$$

$$\dot{m}_7 h_7 + \dot{m}_F h_F + \dot{m}_{FP} h_{FP} = \dot{m}_8 h_8 \tag{4.8}$$

$$\dot{m}_6 h_6 + \dot{m}_9 h_9 + \dot{m}_{10} h_{10} + \dot{m}_{11} h_{11} + \dot{m}_{12} h_{12} + \dot{m}_{13} h_{13} = \dot{m}_{14} h_{14} + \dot{W}_{GT} \tag{4.9}$$

$$\dot{m}_{19}h_{19} = \dot{m}_{20}h_{20} + \dot{W}_{LPST} \tag{4.10}$$

The work required to drive the AC was determined from the energy balance in equation 4.7. The shaft work output of the LPST was determined by the energy balance in equation 4.10.

The combustion process at CCPP1 and CCPP2 uses approximately ~100% excess air, producing exhaust gasses with a mass concentration of approximately ~76% nitrogen, ~21% oxygen, ~1.6% carbon dioxide, and ~1.3% water vapor.

The measure of performance for the CC is referred to as combustion efficiency η_{cc} and is defined in equation 4.11. $\dot{Q}$ is the heat released during combustion, and LHV is the lower heating value of methane.

$$\eta_{CC} = \frac{\dot{Q}}{LHV_{CH_4}} \tag{4.11}$$

The overall isentropic efficiency of the AC is defined in equation 4.12, where subscript *s* indicates the isentropic condition.

$$\eta_{AC} = \frac{h_{7s} - h_1}{h_7 - h_1} \tag{4.12}$$

The isentropic efficiency was also calculated for each individual AC stage, where the beginning of a new stage is defined as the point immediately following each air extraction. Extractions occur at AC rows 6, 11, and 13, dividing the AC into four stages. The isentropic efficiency of stage 2, for example, is defined by equation 4.13.

$$\eta_{AC_2} = \frac{h_{3s} - h_2}{h_3 - h_2} \tag{4.13}$$

The overall isentropic efficiency of the GT and LPST is defined in equations 4.15 and 4.16, respectively.

$$\eta_{GT} = \frac{h_9 - h_{14}}{h_9 - h_{14s}} \tag{4.15}$$

$$\eta_{LPST} = \frac{h_{19} - h_{20}}{h_{19} - h_{20s}} \tag{4.16}$$

4.2: Conventional Exergy Analysis

Exergy analyses are based on the Second Law of Thermodynamics. According to Kotas [35], the Second Law of Thermodynamics is required to establish the difference in quality between thermal and mechanical energy, as well as, indicate the directions of spontaneous processes.

The focus of this thesis is on two main types of exergy: chemical exergy and physical, or thermomechanical, exergy. Physical exergy can be broken down into mechanical exergy, which is dependent on system pressure, and thermal exergy, which is dependent on system temperature. According to [36], chemical exergy is a measure of the maximum work output of the working fluid at the dead state temperature and pressure if it were to be brought into complete thermodynamic equilibrium with the chemical composition of the environment.

The first step in the conventional exergy analysis was an exergy balance. According to [37], for a steady state system, the open system exergy balance is presented in equation 4.16.

$$\sum_j \dot{E}_{q,j} + \dot{W}_{cv} + \sum_i \dot{E}_i - \sum_e \dot{E}_e - \dot{E}_D \tag{4.16}$$

The physical flow exergy e_i of any stream i is defined in equation 4.17. The chemical exergy was determined using tabulated values from [37].

$$e_i = (h_i - h_0) - T_0(s_i - s_0) \tag{4.17}$$

The total exergy rate of streams entering the k components is referred to as the exergy fuel $\dot{E}_{F,k}$. The total exergy rate of streams exiting the k component is referred to as the exergy product $\dot{E}_{P,k}$. The rate of exergy destruction $\dot{E}_{D,k}$ illustrates the magnitude and direction of irreversibilities. The exergy destruction rate $\dot{E}_{D,k}$ of any component k is defined in equation #. Exergy destruction is due to irreversibilities within the system boundaries, whereas exergy losses $\dot{E}_{L,k}$ are due to losses that cross the system boundary, such as heat loss to the environment. In this analysis, the exergy losses were not considered at the component level, so the term was equal to zero for all components.

$$\dot{E}_{D,k} = \dot{E}_{F,k} - \dot{E}_{P,k} - \dot{E}_{L,k} \tag{4.18}$$

According to Dincer and Rosen [26], at the system level, the total rate of fuel exergy $\dot{E}_{D,tot}$ is defined in equation 4.19. Exergy losses $\dot{E}_{L,tot}$ are considered at the system level.

$$\dot{E}_{F,tot} = \dot{E}_{P,tot} + \dot{E}_{D,tot} + \dot{E}_{L,tot} \tag{4.19}$$

Although the steam turbines at CCPP1 and CCPP2 are rated for different mass flow rates, they are setup and operated in a similar manner, and the same methodology can be applied to both machines. The steam turbine system consists of several stages: condensing, high pressure expansion, intermediate pressure superheat, intermediate pressure expansion, low pressure superheat, low pressure reheat, and low pressure expansion. Due to limited data for the HRSG, condenser, and other auxiliary equipment, some of these components could not be analyzed. Therefore the effect of the performance of the components in steam turbine subsystem (ST), on the performance of the combustion turbine system, could not be found. For this reason, when splitting the exergy destruction, the CT and ST were analyzed separately. While the CT and ST do not directly affect one another in this analysis, they both affect the overall power plant.

The exergy balance for a three pressure steam turbine system is presented in equation 4.20. Similar to the gas turbine system, exergy losses were only considered at the system level.

$$\dot{E}_{F,ST} - \dot{E}_{L,ST} - \dot{E}_{P,ST} = \dot{E}_{D,HPST} + \dot{E}_{D,IPST} + \dot{E}_{D,LPST} \tag{4.20}$$

Additional mass and energy is brought into the system by the reheat and superheat processes. This additional work potential was accounted for in the total fuel exergy, $\dot{E}_{F,ST}$, as shown in equation 4.21. The reheat and superheat processes were assumed to operate ideally, therefore their exergy destruction was always equal to zero. The total ST product exergy is presented in equation 4.22.

$$\dot{E}_{F,ST} = \dot{E}_{15} + \dot{E}_{IPRH} + \dot{E}_{LPRH} \tag{4.21}$$

$$\dot{E}_{P,ST} = \dot{E}_{20} + \dot{W}_{HPST} + \dot{W}_{IPST} + \dot{W}_{LPST} \tag{4.22}$$

The exergetic efficiency ε_k is based on work potential, which depends on the quality of energy, as well as the magnitude, making it a more realistic measure of performance than isentropic efficiency. The exergetic efficiency is always greater than the isentropic efficiency. According to Tsatsaronis [38], the exergetic efficiency of component k, is defined in equation 4.23 as the ratio of product and fuel exergies of the component. The exergetic efficiency can be used to compare similar components within a system, and in different systems.

$$\varepsilon_k = \frac{\dot{E}_{P,k}}{\dot{E}_{F,k}} = 1 - \frac{\dot{E}_{D,k}}{\dot{E}_{F,k}} \tag{4.23}$$

The exergy destruction ratio y_k is the only variable in the conventional analysis that can be used to compare dissimilar components within the system. As shown in equation 4.24, it is the ratio of the exergy destruction rate of a single component to all of the exergy inputs in the entire system.

$$y_k = \frac{\dot{E}_{D,k}}{\dot{E}_{F,tot}} \tag{4.24}$$

4.3: Advanced Exergy Analysis

The advanced exergy analysis, as presented in [2, 3] was devised to overcome the limitations of conventional methods. They highlight one method of advanced exergy analysis, in particular, that can be applied to simple and complex thermal systems, the engineering or graph method. The advanced exergy analysis separates itself from other exergy-based methods by answering the questions proposed in [21]. It allows the engineer to quantify the realistically avoidable exergy destruction, and determine if it is due to component interactions or component inefficiencies.

The engineering method requires that the exergy destruction be split multiple times into various components. Splitting the exergy destruction required a model of the plant be made to run the necessary simulations. To complete this task, Engineering Equation Solver (EES) was used. The operational data for the power plants at several points of steady-state operation was input into EES to find the enthalpy, entropy and all other necessary variables. With these properties, the exergy destruction was found for the major components in the model.

4.3.1: Avoidable and Unavoidable Exergy Destruction

The first split of the exergy destruction rate for component k, into its avoidable and unavoidable parts, is defined in eq. (4.25).

$$\dot{E}_{D,k} = \dot{E}_{D,k}^{AV} + \dot{E}_{D,k}^{UN} \tag{4.25}$$

The unavoidable exergy destruction $\dot{E}_{D,k}^{UN}$ is due to technical limitations, and was estimated from the performance of the best available version of the component. If the component was upgraded to the highest performance model available, the remaining avoidable exergy destruction would go to zero.

Using the method outlined in [19], the unavoidable exergy destruction $\dot{E}_{D,k}^{UN}$ was determined by considering each component in isolation, and under the most favorable operating conditions. The unavoidable destruction to production ratio $(\dot{E}_D/\dot{E}_P)_k^{UN}$ was determined by applying the unavoidable assumptions in table 4.2, and calculating the isolated component's unavoidable exergy destruction and exergy product. From equation 4.26, the unavoidable exergy destruction rate of k was then determined [2,3].

$$\dot{E}_{D,k}^{UN} = \dot{E}_{P,k}\left(\frac{\dot{E}_D}{\dot{E}_P}\right)_k^{UN} \tag{4.26}$$

Component	Ideal Conditions	Unavoidable Conditions
Air Compressor (AC)	η_{AC}=100%	η_{AC}=95%
Combustion Chamber (CC)	Q_{loss}=0 ΔP=0	Q_{loss}=0 ΔP=0 λ=.9 T_{in}=2400 ˚F
Expander (GT)	η_{GT}=100%	η_{GT}=95%
Steam Turbines (HPST, IPST, LPST)	η_{ST}=100%	η_{ST}=95%

Table 4.2: Assumptions for ideal and unavoidable conditions

According to [28], the unavoidable exergy destruction ratio in the CC can be found by choosing the lowest technically meaningful value for the air-fuel ratio, and a high inlet air temperature. Altering these parameters as such leads to an increase in performance, and an increase in adiabatic flame temperature T_{ad}. To determine $(\dot{E}_D/\dot{E}_P)_k^{UN}$ for the combustion chamber, the component was considered in isolation from the system, and EES was used calculate T_{ad}.

For the air compressors and expanders, the unavoidable exergy destruction ratio was found by analyzing the component in isolation, with the maximum technically possible values of isentropic efficiency and pressure ratio. As the GT blade cooling additions and extractions have a negative effect on performance, the turbomachines were simulated without them.

The exergetic efficiency was also resolved into its avoidable component, and is defined in equation 4.27.

$$\varepsilon_k^{AV} = \frac{\dot{E}_{P,k}}{\dot{E}_{F,k} - \dot{E}_{D,k}^{UN}} = 1 - \frac{\dot{E}_{D,k}^{AV}}{\dot{E}_{F,k} - \dot{E}_{D,k}^{UN}} \tag{4.27}$$

4.3.2: Endogenous and Exogenous Exergy Destruction

The next step was to split the exergy destruction into its endogenous and exogenous components, as defined in equation 4.28. The endogenous exergy destruction of the k-th component is only due to irreversibilities in the k-th component itself, with all other components acting as ideal. The exogenous exergy destruction of the k-th component is due to irreversibilities caused by other components, component interactions, and the system design.

$$\dot{E}_{D,k} = \dot{E}_{D,k}^{EN} + \dot{E}_{D,k}^{EX} \tag{4.28}$$

In order to find the endogenous exergy destruction of any component k in the system, simulations were run in EES in which the exergy destruction rate of all other components was equal to zero. Various methods including the mass balance and engineering method from [2, 3, 19] were applied to the system unsuccessfully. Eventually, using the method of hybrid processes outlined in [39], meaningful results were found.

The exogenous exergy destruction of component k, $\dot{E}_{D,k}^{EX}$, was determined by equation 4.28. If the endogenous destruction is due solely to irreversibilities within the component itself, it can be assumed that the remaining destruction, $\dot{E}_{D,k}^{EX}$, is due solely to the effects of other components on k.

The two most common ways to reduce the exergy destruction of the CC are to either increase the specific exergy of the products by increasing the temperature of the products, or decrease the specific exergy of the inlet air and fuel by decreasing the temperature of the air and fuel. However, the method used to find the endogenous destruction of the upstream AC requires the inlet temperature of the CC remain constant. A similar situation occurs when analyzing the endogenous destruction of the GT, as the properties of the air entering the transition steam cooler must remain constant. For this situation, the CT subsystem was split into two hybrid processes [39]. To find the endogenous exergy destruction $\dot{E}_{D,k}^{EN}$ of the combustion chamber, work was held constant. The AC was simulated under ideal operating conditions (ε_{AC}=1), with a fixed inlet temperature, leading to a reduction in CC inlet air temperature, and the GT was simulated under ideal operating conditions (ε_{GT}=1), leading to decrease in its exhaust temperature.

To reduce the exergy destruction in either of the ST stages, the same method was used as in the GT. Since there was no combustion process in this system, the temperature and pressure of all streams were held constant, along with the work outputs. Any additional mass brought in by the superheater, reheater, and transition steam cooler was held constant, and only the main stream, $\dot{m}_{15}$, was reduced, resulting in the reduction of the mass flow rate of all additional streams.

4.3.3: Splitting the Endogenous and Exogenous Exergy Destruction

The endogenous and exogenous components of exergy destruction were further split into their avoidable and unavoidable components. The unavoidable component of the endogenous exergy destruction $\dot{E}_{D,k}^{UN,EN}$ was found using equation 4.29.

$$\dot{E}_{D,k}^{UN,EN} = \dot{E}_{D,k}^{EN}\left(\frac{\dot{E}_D}{\dot{E}_P}\right)_k^{UN} \tag{4.29}$$

The unavoidable, exogenous exergy destruction rate of each component was determined by equation 4.30.

$$\dot{E}_{D,k}^{UN,EX} = \dot{E}_{D,k}^{UN} - \dot{E}_{D,k}^{UN,EN} \tag{4.30}$$

The avoidable endogenous and exogenous exergy destruction rate of each component was determined by equations # and #, respectively.

$$\dot{E}_{D,k}^{AV,EN} = \dot{E}_{D,k}^{EN} - \dot{E}_{D,k}^{UN,EN} \tag{4.31}$$

$$\dot{E}_{D,k}^{AV,EX} = \dot{E}_{D,k}^{EX} - \dot{E}_{D,k}^{UN,EX} \tag{4.32}$$

The exergetic efficiency for the avoidable endogenous exergy destruction was determined by equation 4.33.

$$\varepsilon_k^{EN,AV} = 1 - \frac{\dot{E}_{D,k}^{EN,AV}}{\dot{E}_{F,k} - \dot{E}_{D,k}^{UN} - \dot{E}_{D,k}^{EX,AV}} \tag{4.33}$$

4.4: Conventional Exergoeconomic Analysis

4.4.1: Investment Costs

The first step in the exergoeconomic analysis was to calculate the investment costs I_k of each component. The investment cost equations are defined for the AC, CC, and GT in equations 4.36, 4.37, and 4.38, respectively. Unfortunately, it was unclear in how to determine the constants, C_{11}-C_{34}, for systems other than those in [2, 25, 36, 40]. Bejan et al. [36] made the assumption that the same constants can be applied to all of their examples, but noted that for a real system this was untrue. From trial and error, it was clear that their constants could not be used for CCPP2.

According to [25], the cost information which is assumed in the analysis is never available. Even if a quote was given for the exact system specifications, it would not indicate the cost of the individual components, as units at this capacity are priced as a package. In [37], Tsatsaronis and Cziesla indicated that when a vendor quote or professional cost estimate is not available, the purchase costs can be estimated from the literature. The investment cost I_k for each component was estimated by applying the same ratio of equipment costs indicated in [25] to the cost of a simple-cycle combustion gas turbine system comparable to the CT at CCPP2 [41]. These estimates are indicated in appendix A-1.

$$I_{AC} = \frac{C_{11}\dot{m}_a}{C_{12} - \eta_{AC}} \frac{P_{out}}{P_{in}} \ln\left(\frac{P_{out}}{P_{in}}\right) \tag{4.34}$$

$$I_{CC} = \frac{C_{21}\dot{m}_a}{C_{22} - \frac{P_{out}}{P_{in}}} \left[\exp(C_{23}T_{out} - C_{24})\right] \tag{4.35}$$

$$I_{GT} = \frac{C_{31}\dot{m}_{gas}}{C_{32} - \eta_{GT}} \ln\left(\frac{P_{in}}{P_{out}}\right) \left[\exp(C_{33}T_{in} - C_{34})\right] \tag{4.36}$$

The estimated I_k was input into equation 4.37 to find the levelized annual cost rate associated with the capital investment and maintenance costs. Where β is the capital recovery factor (β=18.2%), γ is the operating and maintenance costs (γ=1.092%), and τ is the annual operating hours of the plants (τ=8000 hrs). Levelized annual values were used in the analysis as the cost components in a thermal system can vary drastically over its economic life [37].

$$\dot{Z}_k = (\beta + \gamma) \frac{I_k}{\tau} \tag{4.37}$$

4.4.2: Exergy Costs

Knowledge of the cost of exergy destruction in a component is a very useful parameter for improving the cost-effectiveness of a system. In thermoeconomic analysis [37], the cost rates associated with each exergy stream are used to calculate thermoeconomic variables for the components. Minimizing the cost of a thermal system involves find the optimal trade-off between the cost rates of exergy and capital investment.

The cost rate of any exergy stream j is defined in equation 4.38, where $\dot{E}_j$ is the stream exergy rate and c_j is the average cost per unit exergy in ($/MMBtu).

$$\dot{C}_j = \dot{E}_j \cdot c_j \tag{4.38}$$

The cost of exergy associated with heat and work is defined in equations 4.39 and 4.40, respectively.

$$\dot{C}_q = c_q \cdot \dot{Q} \tag{4.39}$$

$$\dot{C}_w = c_w \cdot \dot{W} \tag{4.40}$$

Cost balances for each component, and any necessary auxiliary cost equations, were used to determine the values for c_j, c_w, and c_q. The cost balance for any component k is defined in equation 4.41, accounting for the exergy entering and exiting the component, as well as the component's investment cost.

$$\sum_{j=1}^{n} \dot{C}_{j,k,in} + \dot{Z}_k = \sum_{j=1}^{m} \dot{C}_{j,k,out} \tag{4.41}$$

As the cost of entering exergy streams was unknown, auxiliary cost equations were devised for components in which the number of entering and exiting streams were not equal. General cost balances for thermal system components can be found in [37]. The exact cost equations used to find all of the unknowns at CCPP2 can be found in the EES code in appendix A-2.

Bejan et al. [36] explains that there are two ways to determine the cost of the exergy destruction $\dot{C}_{D,k}$ within a component, as illustrated in equations 4.42 and 4.43. If it is assumed that the product $\dot{E}_{P,k}$ is fixed and the unit cost of fuel $c_{F,k}$ is independent of the exergy destruction, equation 4.42 should be used. Alternatively, if it is assumed that the fuel $\dot{E}_{F,k}$ is fixed and the unit cost of product $c_{P,k}$ is independent of the exergy destruction, equation 4.43 should be used. In reality, $\dot{C}_{D,k}$ would lie somewhere between the two values. In the analysis of CCPP2, the product exergy was fixed, therefore equation 4.44 was used.

$$\dot{C}_{D,k} = c_{F,k}\dot{E}_{D,k} \tag{4.42}$$

$$\dot{C}_{D,k} = c_{P,k}\dot{E}_{D,k} \tag{4.43}$$

According to [37] the cost rate of exergy destruction ($\dot{C}_{D,k}$) is a “hidden” cost that can only be revealed through exergoeconomic analysis. For most components, $\dot{C}_{D,k}$ and the capital investment cost of the component $\dot{Z}_k^{CI}$ have an inverse relationship. For a decrease in exergy destruction, there is a decrease in $\dot{C}_{D,k}$ and an increase in $\dot{Z}_k^{CI}$. For thermoeconomic optimization of the system, the optimal trade-off between $\dot{C}_{D,k}$ and $\dot{Z}_k^{CI}$ is sought after, so as to minimize their sum.

The relative cost difference r_k between the cost per unit exergy of the product and fuel is defined in equation 4.44.

$$r_k = \frac{c_{P,k} - c_{F,k}}{c_{F,k}} = \frac{1 - \varepsilon_k}{\varepsilon_k} + \frac{\dot{Z}_k}{c_{F,k}\dot{E}_{P,k}} \tag{4.44}$$

$\dot{Z}_k$ is equal to the sum of the capital investment cost rate $\dot{Z}_k^{CI}$ and the operations and maintenance cost rate $\dot{Z}_k^{OM}$. However, the cost of operations and maintenance is of little consequence to the optimization of the system, and is here forth neglected. Equation 4.44 indicates that cost sources in a component can be grouped as exergy related and non-exergy related costs. It is important to know the relative significance of each category in evaluating component performance.

The exergoeconomic factor f_k reveals the percent contribution of the capital investment cost to the sum of the capital investment cost $\dot{Z}_k^{CI}$ and cost of exergy destruction $\dot{C}_{D,k}$ of component k, and is defined in equation 4.45.

$$f_k = \frac{\dot{Z}_k^{CI}}{\dot{Z}_k^{CI} + \dot{C}_{D,k}} \tag{4.45}$$

A high value for f_k would suggest that investment cost of the component be reduced, even at the expense of exergetic efficiency. A low value for f_k would suggest that the component may be improved by improving its efficiency, even if capital investment cost would increase.

4.5: Advanced Exergoeconomic Analysis

4.5.1: Splitting the Cost Rate of Capital Investment

Outlined in [2], the method for determining the unavoidable component of the cost rate of capital investment is similar to that of the exergy destruction. For example, in the AC the first step is to identify the lowest operational isentropic efficiency, $\eta_{AC,min}$, then simulate the system with the AC operating at $\eta_{AC,min}$, and all other components operating at their least efficient, while keeping the specific work values constant. The resulting air mass flow rate, as well as $\eta_{AC,min}$ is then input into equation # 4.34 to determine the unavoidable investment cost I_k. Unfortunately, due to the lack of information for the constants of equations 4.34, 4.35, and 4.36, it was not possible to accurately split the investment cost into its endogenous, exogenous, avoidable, or unavoidable components.

4.5.2: Splitting the Cost Rate of Exergy Destruction

While splitting the capital investment costs was not possible, a further analysis was still completed by first splitting the exergy costs rates, as defined in [2]. For the avoidable and unavoidable exergy destruction, cost rates are defined by equations 4.46 and 4.47, respectively. The overall cost rate of exergy destruction is the sum of these values, as defined in equation 4.48.

$$\dot{C}_{D,k}^{AV} = c_{F,k}\dot{E}_{D,k}^{AV} \tag{4.46}$$

$$\dot{C}_{D,k}^{UN} = c_{F,k}\dot{E}_{D,k}^{UN} \tag{4.47}$$

$$\dot{C}_{D,k} = \dot{C}_{D,k}^{AV} + \dot{C}_{D,k}^{UN} \tag{4.48}$$

The cost rates associated with the endogenous and exogenous exergy destruction are defined in equation 4.49 and equation 4.50, respectively. The overall cost rate of exergy destruction is the sum of these values, as presented in equation 4.53.

$$\dot{C}_{D,k}^{EN} = c_{F,k}\dot{E}_{D,k}^{EN} \tag{4.49}$$

$$\dot{C}_{D,k}^{EX} = c_{F,k}\dot{E}_{D,k}^{EX} \tag{4.50}$$

$$\dot{C}_{D,k} = \dot{C}_{D,k}^{EN} + \dot{C}_{D,k}^{EX} \tag{4.51}$$

The cost rates associated with the avoidable and unavoidable endogenous and avoidable, and unavoidable exogenous components of exergy destruction, are defined in equations 4.52 through 4.55, respectively.

$$\dot{C}_{D,k}^{EN,AV} = c_{F,k}\dot{E}_{D,k}^{EN,AV} \tag{4.52}$$

$$\dot{C}_{D,k}^{EN,UN} = c_{F,k}\dot{E}_{D,k}^{EN,UN} \tag{4.53}$$

$$\dot{C}_{D,k}^{EX,AV} = c_{F,k}\dot{E}_{D,k}^{EX,AV} \tag{4.54}$$

$$\dot{C}_{D,k}^{EX,UN} = c_{F,k}\dot{E}_{D,k}^{EX,UN} \tag{4.55}$$

In [2], the exergoeconomic factor was also resolved into its avoidable endogenous component. This was done to illustrate the percent contribution of the avoidable endogenous capital investment cost $\dot{Z}_k^{EN,AV}$ to the sum of $\dot{Z}_k^{EN,AV}$ and the cost of avoidable endogenous exergy destruction $\dot{C}_{D,k}^{EN,AV}$ of component k. However, because the cost associated with capital investment could not be split, the equation for the avoidable endogenous exergoeconomic factor $f_k^{EN,AV}$ was modified for use in this analysis. The modified avoidable endogenous exergoeconomic factor $f_k^{EN,AV*}$ is defined in equation 4.56.

$$f_k^{EN,AV\star} = \frac{\dot{Z}_k^{CI}}{\dot{Z}_k^{CI} + \dot{C}_{D,k}^{EN,AV}} \tag{4.56}$$

5. RESULTS AND DISCUSSION

5.1: Energy Analysis

While the main focus of this thesis was the on 2^{nd} Law of Thermodynamics, 1st Law relations were useful for comparing the relative performance between similar components at the two plants. For the 1^{st} Law (energy) analysis, two weeks of operational data was selected, with 10 minute intervals. The weeks chosen were those in which both power plants were in fully-functional operation. Using a large amount of data gave a good representation of the average performance of components during a variety operating conditions. The data was filtered using Excel to eliminate times when evaporative cooling, anti-icing, and steam injection was present, as these auxiliaries affect the power plant performance.

Due to non-disclosure agreements, any identifiable information for the power plants, in addition to the source of the data could not be disclosed. The results were plotted using Excel with second degree polynomial trend lines. The coefficient of determination, R^2, for each trend line is denoted in the upper right hand corner of each plot. The fits were good, however, this method was only used as an illustration of certain relationships.

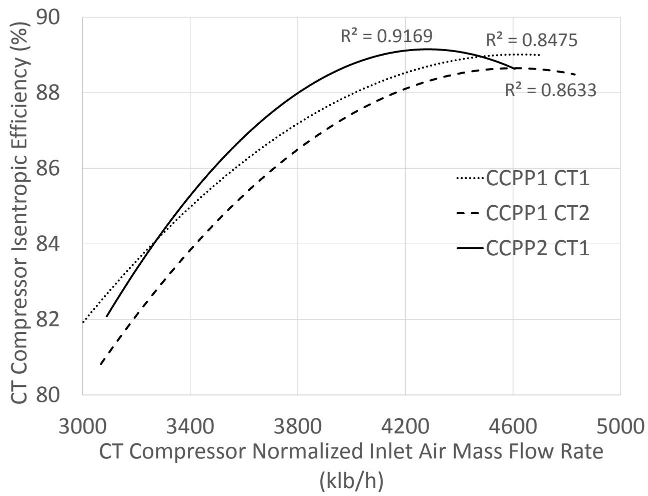

Figure 5.1: Relationship between AC isentropic efficiency and normalized inlet air mass flow rate

The compressor inlet air mass flow rate is a useful parameter for comparison of similar air compressors (AC), as it is proportional to the net power output of the combustion turbine (CT). To ensure consistent inlet conditions for each case, the AC inlet air mass flow rate was normalized with respect to temperature and static pressure. Figure 5.1 illustrates the relationship between compressor isentropic efficiency and the normalized inlet air mass flow rate, for each AC. It is evident that an increase in mass flow rate resulted in an increase in isentropic efficiency, with a polynomial trend. In all three compressors, at high flow rates, there was a point at which the efficiency peaked and started to decrease.

At CCPP1, it was evident that the CT1 AC had a higher isentropic efficiency than the CT2 AC over all normal operating loads. Since the two gas turbines, and their operating parameters were nearly identical, it was assumed that the difference in performance was due to factors such as uneven wear and fouling, measurement equipment uncertainty, and inherent variations in the equipment. The CCPP2 AC exhibited better performance than either AC at CCPP1. The trends of the three units were of similar shape, with only slight variations, which could be due to factors such as a different configuration, or area, of the AC inlet ductwork.

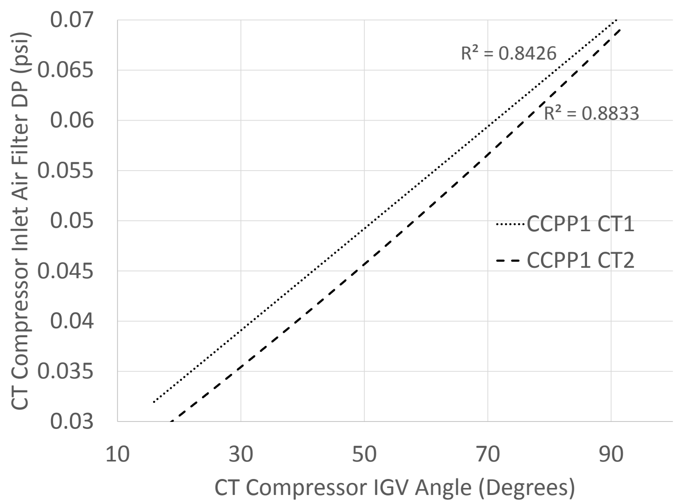

Figure 5.2: Relationship between CT compressor inlet air filter differential pressure and IGV angle

Buildup of dirt and debris on the AC inlet air filter results in a larger-than-normal pressure differential across the filter, and can have a noticeable effect on AC performance. By measuring the pressure differential across the filter, the level of buildup was estimated. Figure 5.2 illustrates the relationship between the AC inlet air filter differential pressure and IGV angle. A larger IGV angle corresponds to greater unit power output. The CCPP1 CT1 AC filter appears to have a slightly larger differential pressure, implying there was slightly greater buildup than in CT2, however the variation between the two was negligible. Knowledge of this relationship is useful when diagnosing the sources of performance variation between identical units, as the only difference may be a dirty filter. Unfortunately data was not available for the CCPP2 air filter.

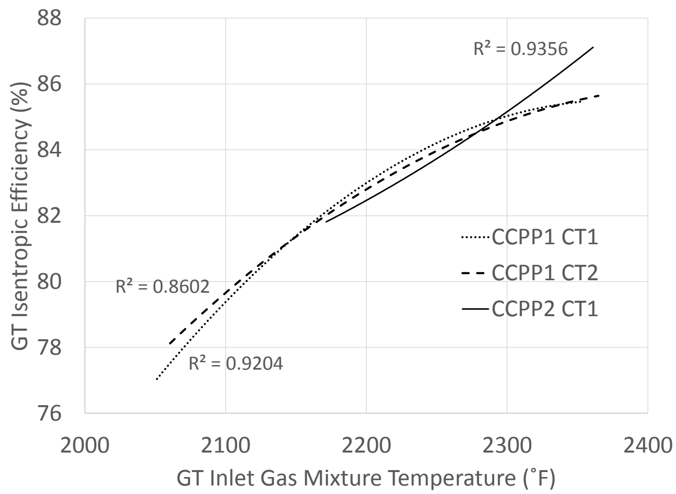

Figure 5.3: Relationship between GT inlet gas mixture temperature and GT isentropic efficiency

The relationship between the GT inlet gas mixture temperature and isentropic efficiency for the three expanders is presented in figure 5.3. The GT inlet gas mixture temperature is the mixture temperature of the combustion exhaust gasses and the first stage of cooling air from the rotor air cooler. There was clearly a strong correlation between the two parameters, with a greater inlet temperature corresponding to a greater isentropic efficiency. At CCPP1, both expanders had similar performance over the range of operation. It was evident that variation existed between the expanders at the different plants, as the shapes of the trend lines differ. This variation is most likely due to different schedules for the use of cooling air at the different plants, as it is ultimately the decision of the plant operator to determine the cooling air schedule. A reduction in cooling air leads to an increase in GT inlet gas mixture temperature and, consequently, an increase in GT performance.

In the next section, 5.2, a single point of operation was chosen at which to evaluate the power plant components. Table 5.1 presents the parameters determined in the 1st Law evaluation for that point, which were necessary in order to conduct the 2nd Law analysis.

Component	$\dot{W}_k$ (MW)	η_k (%)
AC	260.65	88.50
GT	463.95	87.55
HPST	22.35	77.17
IPST	32.58	61.69
LPST	59.95	90.75

Table 5.1: 1st Law parameters for the evaluated point of operation at CCPP2

5.2: Conventional Exergy Analysis

For the 2nd Law (exergy) analysis, a single point in of operation was carefully chosen from the two weeks of operational data, and was considered to be a good representation of the plant's average performance. The exergetic properties of all of streams under investigation are presented in table 5.2, and were used in the following exergy analysis.

Stream	Fluid	$\dot{m}$ (klb/h)	T (˚F)	P (psia)	e^{ph} (Btu/lb)	e^{ch} (Btu/lb)	$\dot{E}_{j,total}$ (MW)
1	Air	4800.00	51.10	14.39	0.00	0.00	0.00
2	Air	120.20	400.20	70.76	75.95	0.00	2.68
3	Air	9.83	639.90	141.50	128.60	0.00	0.37
4	Air	154.00	728.90	212.30	155.20	0.00	7.00
5	Air	132.20	816.10	283.00	178.40	0.00	6.91
6	Air	132.20	405.00	283.00	125.00	0.00	4.84
7	Air	4384.00	816.10	283.00	178.40	0.00	229.22
F	Methane	105.20	436.00	581.80	296.30	22089.40	690.20
FP	Methane	4.38	56.75	581.80	250.30	22089.40	28.66
8	Gas Mix	4494.00	2425.00	283.00	548.06	9.54	734.42
9	Gas Mix	4494.00	2407.00	283.00	541.16	9.54	725.34
mix	Gas Mix	4626.20	2284.00	283.00	496.06	9.54	685.52
10	Air	154.00	575.30	212.30	134.30	0.00	6.06
11	Air	9.83	639.90	141.50	128.60	0.00	0.37
12	Air	5.53	400.20	70.76	75.95	0.00	0.12
13	Air	114.60	400.20	70.76	75.95	0.00	2.55
14	Gas Mix	4910.00	1075.00	15.64	127.76	9.54	197.58
15	Steam	537.10	997.00	1726.00	646.00	214.45	135.45
16	Steam	537.10	657.80	401.40	483.10	214.45	109.80
17	Steam	614.80	970.20	380.80	578.50	214.45	142.88
18	Steam	614.80	586.40	32.43	333.30	214.45	98.70
19	Steam	700.60	577.90	32.43	331.20	214.45	112.04
20	Steam	700.60	107.50	1.12	66.10	214.45	57.61

Table 5.2: Exergetic properties of streams in the system

In table 5.3, the results of the conventional 2nd Law (exergy) analysis for the major components are presented in order of decreasing magnitude of exergy destruction. The results will be discussed in this order as well.

Component	$\dot{E}_{F,k}$ (MW)	$\dot{E}_{P,k}$ (MW)	$\dot{E}_{D,k}$ (MW)	ε_k (%)	y_k (%)
CC	948.40	734.40	214.00	77.44	29.77
GT	694.64	661.50	33.14	95.23	4.61
AC	260.65	246.20	14.45	94.46	2.01
IPST	104.25	92.67	11.58	88.89	8.55
LPST	68.00	57.75	10.25	84.93	7.57
HPST	101.68	98.43	3.25	96.80	2.40

Table 5.3: Results of the conventional exergy analysis

Consistent with previous studies [2,3,11,16,19], the combustion chamber had the largest value of exergy destruction $\dot{E}_{D,k}$ among the components analyzed, with 214 MW, as well as, the lowest exergetic efficiency ε_k, of 77.44%. Compared to the range of combustion chamber exergetic efficiencies between 62 and 80% in [2,3,19,25,36], its exergetic performance was excellent. However, 74.65% of the exergy destruction among the components analyzed in the CCPP2 system occurred in the CC.

The second largest source of exergy destruction in the system was the expander (GT), with 33.14 MW. With an exergetic efficiency of 95.23%, it had average exergetic performance relative to other gas turbines in the literature (93-97%) [2,3,19,25,36]. The HPST, with 3.25 MW of destruction and an exergetic efficiency of 96.8%, was the best performing component in the conventional exergy analysis.

While the results of the conventional exergetic analysis revealed that there was potential for improvement, there was still much to be desired. The most effective means by which to alter the system, the maximum possible decrease of exergy destruction within each system component, and the effect of a single component's performance on the exergy destruction of other components were still unknown.

5.3: Advanced Exergy Analysis

5.3.1: Avoidable and Unavoidable Exergy Destruction

In table 5.4, the results of the splitting of exergy destruction of the k-th component into its unavoidable and avoidable components are presented in order of decreasing magnitude of avoidable exergy destruction $\dot{E}_{D,k}$. The plant components will be discussed in the same order. The unavoidable destruction to production ratio, $(\dot{E}_D/\dot{E}_P)_k^{UN}$ was determined by applying the unavoidable assumptions to each component in isolation.

Component	$\dot{E}_{D,k}$ (MW)	$\dot{E}_{P,k}$ (MW)	$(\frac{\dot{E}_D}{\dot{E}_P})_k^{UN}$	$\dot{E}_{D,k}^{UN}$ (MW)	$\dot{E}_{D,k}^{AV}$ (MW)	ε_k^{AV} (%)	$x_{D,k}^{AV}$ (%)
CC	214.00	734.40	0.2295	168.54	45.46	94.17	21.24
GT	33.14	661.50	0.0266	17.58	15.56	97.70	46.94
IPST	11.58	92.67	0.0244	2.26	9.32	90.86	80.50
AC	14.45	246.20	0.0217	5.35	9.10	96.44	62.98
LPST	10.25	57.75	0.0351	2.03	8.22	87.54	80.22
HPST	3.25	98.43	0.0189	1.86	1.39	98.61	42.67

Table 5.4: Avoidable and unavoidable exergy destruction in CCPP2 components

The CC had the largest magnitude of avoidable exergy destruction, with 45.46 MW. Therefore, 45.46 MW of exergy destruction could potentially be avoided if the component was improved. This could be achieved by reducing the CC air-fuel ratio, and further preheating the combustion air. However, the operating conditions of the CC are limited by the other components of the system, and environmental regulations for the concentration of pollutant species in the exhaust gasses. In lean-premixed combustion, as in CCPP2, the combustion chamber has a minimum excess air requirement to keep the

NOx and SOx levels in the exhaust gasses within regulation**.** Emissions can also be reduced by means such as gas scrubbing and chemical injection, allowing for the CC to be run with less excess air [42].

Although the CC had the largest magnitude of avoidable destruction, its fraction of avoidable exergy destruction $x_{D,k}^{AV}$ of 21.24%, was the lowest in the system, indicating that it might not be the most effective component to improve first. A low fraction of avoidable exergy destruction indicates that the component is performing almost as well as technically possible, and therefore has less potential for improvement.

The expander had the second largest magnitude of avoidable exergy destruction, 15.56 MW, suggesting it be considered for improvement second, but a fraction of avoidable exergy destruction of only 46.94%. This result suggests that 15.56 MW of destruction in the GT can be avoided by upgrading the component to the highest performance version available (η=95%), and running the turbine without cooling air.

While the AC, IPST, and LPST had similar magnitudes of avoidable destruction, the destruction in the IPST and LPST was 80.5% and 80.22% avoidable, respectively, whereas less than half of the destruction in the AC was avoidable. With a greater potential for improvement, improving the IPST and LPST performance should be of higher priority than improving the AC. With an avoidable exergy destruction of merely 1.39 MW, and the lowest percentage of avoidable exergy destruction, improving the HPST would seem to have little effect on the performance of the system.

5.3.2: Endogenous and Exogenous Exergy Destruction

Due to the limitations mentioned in section 4.2.1, the combustion turbine and steam turbine were considered as separate systems from this section onwards. In table 5.4, the results of the splitting of exergy destruction into its endogenous $\dot{E}_{D,k}^{EN}$ and exogenous $\dot{E}_{D,k}^{EX}$ components are presented for the combustion turbine system in order of decreasing magnitude of endogenous exergy destruction.

Component	ε_k (%)	$\dot{E}_{D,k}$ (MW)	$\dot{E}_{D,k}^{EN}$ (MW)	$\dot{E}_{D,k}^{EX}$ (MW)	$x_{D,k}^{EN}$ (%)
CC	77.44	214.00	130.00	84.00	60.75
GT	95.23	33.14	19.43	13.71	58.63
AC	94.46	14.45	8.85	5.60	61.25

Table 5.5: Endogenous and exogenous exergy destruction in CCPP2 CT components

The results of splitting the exergy destruction into its endogenous and exogenous components revealed that CC had the largest magnitude of endogenous exergy destruction, with 130 MW. Therefore, as the AC, CC, and GT all had similar fractions of endogenous destruction, the results indicate that it would be most beneficial to improve the CC first, followed by the GT then AC.

With an exogenous exergy destruction of 84 MW for the CC, for example, there is also significant potential for improvement by improving other components in the system. The effect of the individual component's destruction on one another would need to be quantified in order to determine the significance, however. These values were quantified in [19], by adding the k-th component's avoidable endogenous destruction with the exogenous exergy destruction of other components due to the k-th component.

They did not find the additional reduction in system exergy destruction significant enough to affect the suggested order of improvement of the components, however the reduction was worth noting. Further splitting the endogenous destruction into avoidable and unavoidable components, as done in section 5.3.3, seek to reveal the amount of exergy destruction which can realistically be avoided.

Due to the limited amount of data for the HRSG, the accuracy of splitting the endogenous and exogenous destruction of the steam turbine components could not be confirmed. The results of the unavoidable and avoidable parts of the exergy destruction were not affected as each component was isolated from the system during analysis. For accurate results, more information about the other components of the ST system must be known.

5.3.3: Splitting the Endogenous and Exogenous Exergy Destruction

The endogenous exergy destruction of the CT components was further split into its unavoidable and avoidable components, and the results are presented in table 5.6. The components are presented and discussed in order of decreasing magnitude of avoidable endogenous destruction $\dot{E}_{D,k}^{EN,AV}$. The avoidable endogenous exergy destruction quantifies the potential reduction in each respective component's exergy destruction, if it were running under the specified unavoidable conditions. This gives a better view of the realistically avoidable exergy destruction.

Component	$\dot{E}_{D,k}$ (MW)	$\dot{E}_{D,k}^{EN}$ (MW)	$\dot{E}_{D,k}^{EX}$ (MW)	$(\frac{\dot{E}_D}{\dot{E}_P})_k^{UN}$	$\dot{E}_{D,k}^{EN,UN}$ (MW)	$\dot{E}_{D,k}^{EX,UN}$ (MW)	$\dot{E}_{D,k}^{EN,AV}$ (MW)	$\dot{E}_{D,k}^{EX,AV}$ (MW)	$x_{D,k}^{EN,AV}$ (%)	$x_{D,k}^{EX,AV}$ (%)
CC	214.00	130.00	84.00	0.2295	100.75	67.79	29.25	16.21	22.50	19.29
GT	33.14	19.43	13.71	0.0266	11.63	5.97	7.80	7.74	40.15	56.48
AC	14.45	8.85	5.60	0.0217	2.58	2.76	6.27	2.84	70.80	50.75

Table 5.6: Avoidable and unavoidable endogenous exergy destruction for CCPP2 CT

The CC had the largest magnitude of avoidable endogenous destruction $\dot{E}_{D,k}^{EN,AV}$, with 29.25 MW, however it also had the lowest fraction of avoidable endogenous destruction $x_{D,k}^{EN,AV}$ with 22.5%. The GT had the second largest magnitude of $\dot{E}_{D,k}^{EN,AV}$ with 7.8 MW, and a fraction of avoidable endogenous destruction $x_{D,k}^{EN,AV}$ of 40.15%. While the AC had the lowest magnitude of avoidable endogenous exergy destruction, 70.8% of its endogenous destruction was avoidable. Over 50% of the exogenous exergy destruction in the AC and GT was avoidable, which indicated significant potential for improvement by improving component interactions.

The significant difference between the quantifiable results of the conventional and advanced exergy analyses highlights the usefulness of splitting the destruction to find the avoidable endogenous component. For example, with only the results of section 5.3.1, it might seem as though upgrading the GT would result in up to a 33.13 MW reduction in exergy destruction, or upgrading the CC would result in up to a 214 MW reduction. However, from further analysis, it became evident that only a maximum reduction of 7.8 MW in the GT and 29.25 MW in the CC were possible when upgrading the respective component.

5.4: Exergoeconomic Analysis

5.4.1: Conventional Exergoeconomic Analysis

Until now, considerations for which component to improve, as well as, how the improvement could be made, were based solely on thermodynamics. The exergoeconomic analysis to follow was used to make those decisions based on cost. While thermodynamic performance is important, the importance of the economics is more significant.

The CT subsystem of CCPP2 contributed 64% of the overall power production in the plant, and over 91% of the exergy destruction among the components analyzed. The ST subsystem contributed 36% of the overall power production, and less than 9% of the exergy destruction among the components analyzed. Due to the lack of contribution by the ST subsystem, focus was placed on evaluating and improving only the CT subsystem through exergoeconomic analysis and optimization. It must be emphasized that these percentages of exergy destruction are only for the components under investigation. If more information was available for the HRSG, and all components in the power plant were evaluated, this would not be the case.

In table 5.7, results for the conventional exergoeconomic analysis are presented in in order of decreasing sum $\dot{Z}_k^{CI} + \dot{C}_{D,k}$. The importance of this parameter was discussed in section 4.4.2.

Component	I_k (10^3 \$)	ε_k (%)	$\dot{E}_{D,k}$ (MW)	$c_{F,k}$ (\$/MMBtu)	$c_{P,k}$ (\$/MMBtu)	$\dot{C}_{D,k}$ (\$/h)	$\dot{Z}_k^{CI}$ (\$/h)	r_k (%)	f_k (%)	$\dot{Z}_k^{CI}+\dot{C}_{D,k}$ (\$/h)
CC	1988.32	77.44	214.00	8.96	12.84	1914.00	45.51	43.35	2.32	1959.51
GT	26037.74	95.23	33.14	16.30	22.22	540.30	595.80	36.32	52.44	1136.10
AC	18218.56	94.46	14.45	22.22	23.25	321.20	417.00	4.64	56.49	738.20

Table 5.7: Results of the conventional exergoeconomic analysis

With the largest $\dot{Z}_k^{CI} + \dot{C}_{D,k}$ of 1959.51 ($/h), the results of the conventional analysis for the CT system indicate that emphasis should be placed on first improving the performance of the CC, followed by the GT, and then the AC. A very low value of the exergoeconomic factor f_k indicated that most of the costs in the CC were associated with exergy destruction rather than capital investment. Therefore, it would be suggested that the efficiency of the CC be improved, even if it leads to an increase in capital investment cost. Methods for increasing the efficiency of the CC were discussed in section 5.3.1. The impact of changes in operating parameters on the levelized capital investment cost $\dot{Z}_k$ can be determined by equations 4.34 through 4.36 in methodology.

The AC and GT are said to have target values of f_k between 25% and 65% [37]. The AC and GT at CCPP2, f_k values around 50% indicated that there was a fairly even trade-off between cost of capital investment and cost of exergy destruction. Since these parameters are inversely related, a better trade-off could prove very difficult to find. Therefore, from the results of the conventional thermoeconomic analysis, no recommendations can be made for improving the AC or GT. Since the conventional analysis does not consider the technical limitation of reducing the exergy destruction, to investigate further, the cost rates were split in the advanced exergoeconomic analysis.

5.4.2: Advanced Exergoeconomic Analysis

In the advanced exergoeconomic analysis, the cost rate of exergy destruction was split into its avoidable, unavoidable, endogenous, exogenous, avoidable endogenous, unavoidable endogenous, avoidable exogenous, and unavoidable exogenous components for the each component in the CT system. The results of splitting the cost of exergy destruction are presented in table 5.8 in order of decreasing cost of avoidable endogenous exergy destruction $\dot{C}_{D,k}^{EN,AV}$.

Component	$\dot{C}_{D,k}$ ($/h)	$\dot{C}_{D,k}^{AV}$ ($/h)	$\dot{C}_{D,k}^{UN}$ ($/h)	$\dot{C}_{D,k}^{EN}$ ($/h)	$\dot{C}_{D,k}^{EX}$ ($/h)	$\dot{C}_{D,k}^{EN,AV}$ ($/h)	$\dot{C}_{D,k}^{EN,UN}$ ($/h)	$\dot{C}_{D,k}^{EX,AV}$ ($/h)	$\dot{C}_{D,k}^{EX,UN}$ ($/h)
CC	1914	407.2	1510	1164	752.4	262	902.4	145.2	607.2
AC	321.1	202.2	118.9	196.60	124.40	139.3	57.33	63.10	61.33
GT	540.3	253.7	286.6	316.8	223.5	127.2	189.6	126.2	97.33

Table 5.8: Results of splitting the cost of exergy destruction in the CT components

The most important parameters for the conventional and advanced exergoeconomic analyses are presented in table 5.9 for comparison. Components are presented in order of decreasing sum $\dot{Z}_k^{CI} + \dot{C}_{D,k}^{EN,AV}$.

Component	ε_k (%)	r_k (%)	f_k (%)	$\dot{Z}_k^{CI}+\dot{C}_{D,k}$ ($/h)	$\varepsilon_k^{EN,AV}$ (%)	$f_k^{EN,AV}$ (%)	$\dot{Z}_k^{CI}+\dot{C}_{D,k}^{EN,AV}$ ($/h)
GT	95.23	36.32	52.44	1136.10	98.36	82.41	723.00
AC	94.46	4.64	56.49	738.20	97.52	74.96	556.30
CC	77.44	43.35	2.32	1959.51	96.34	14.80	307.51

Table 5.9: Comparison of the advanced and conventional exergoeconomic analysis for the CT components

Compared to the conventional sum $\dot{Z}_k^{CI} + \dot{C}_{D,k}$, the order of the components had changed. From table 5.8, it became evident that this change was due to the lower cost of avoidable endogenous cost of exergy destruction $\dot{C}_{D,k}^{EN,AV}$ compared to the overall cost of exergy destruction of each component $\dot{C}_{D,k}$. While a little less than half the cost of exergy destruction in the AC was avoidable endogenous, only about a quarter of the cost of exergy destruction in the GT was avoidable endogenous, and only about an eighth of the cost of exergy destruction in the CC is avoidable endogenous.

It should be noted that the avoidable endogenous component of the levelized cost of capital investment must also be found in order to accurately determine the order for improvement based on the advanced exergoeconomic method outlined in [Kelly]. For this reason, as well as the relatively small margin between the values for the sum $\dot{Z}_k^{CI} + \dot{C}_{D,k}^{EN,AV}$ for each component, conclusions for the most cost-effective component to improve could not be drawn from these results.

5.5: Sensitivity Analysis

5.5.1: Sensitivity of the Unavoidable Production to Destruction Ratio

Splitting the exergy destruction of the k-th component into unavoidable and avoidable components required several assumptions be made. Since all avoidable and unavoidable exergy destruction and cost is directly proportional to the value of the unavoidable exergy destruction to production ratio $(\dot{E}_D/\dot{E}_P)_k^{UN}$, its sensitivity was investigated by varying the unavoidable conditions. The sensitivity of $(\dot{E}_D/\dot{E}_P)_k^{UN}$ with respect to isentropic efficiency and pressure ratio are presented for the AC, GT, and the three ST stages in figures 5.4, 5.5, 5.6, 5.7, 5.9, and 5.10, respectively. The sensitivity of $(\dot{E}_D/\dot{E}_P)_k^{UN}$ with respect to the excess air fraction is presented for the CC in figure 5.8.

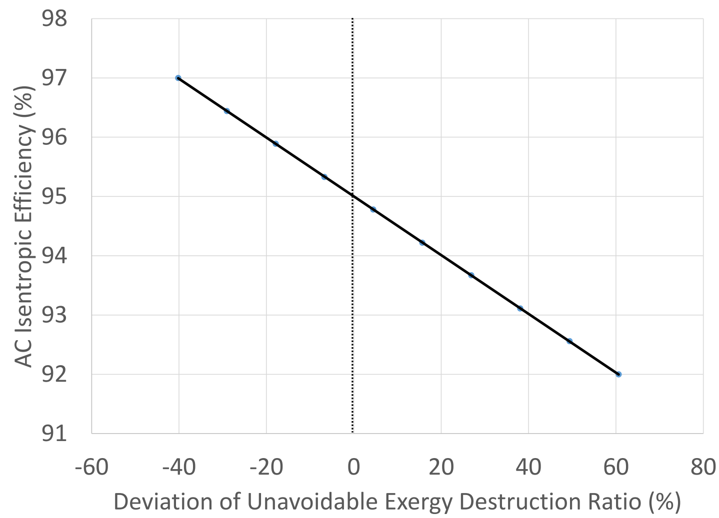

Figure 5.4: Sensitivity of the AC unavoidable exergy destruction ratio to changes in AC isentropic efficiency

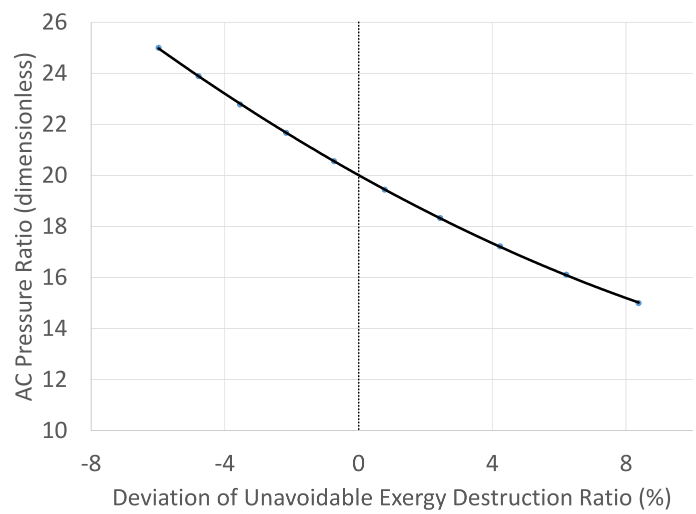

Figure 5.5: Sensitivity of the AC unavoidable exergy destruction ratio to changes in AC pressure ratio

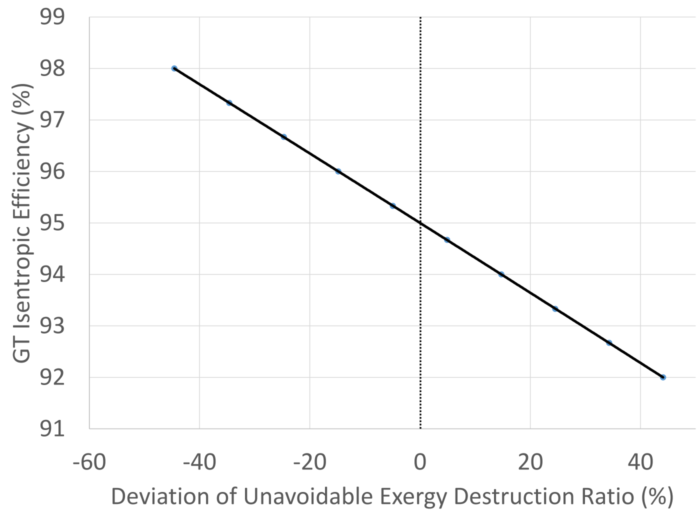

Figure 5.6: Sensitivity of the GT unavoidable exergy destruction ratio to changes in GT isentropic efficiency

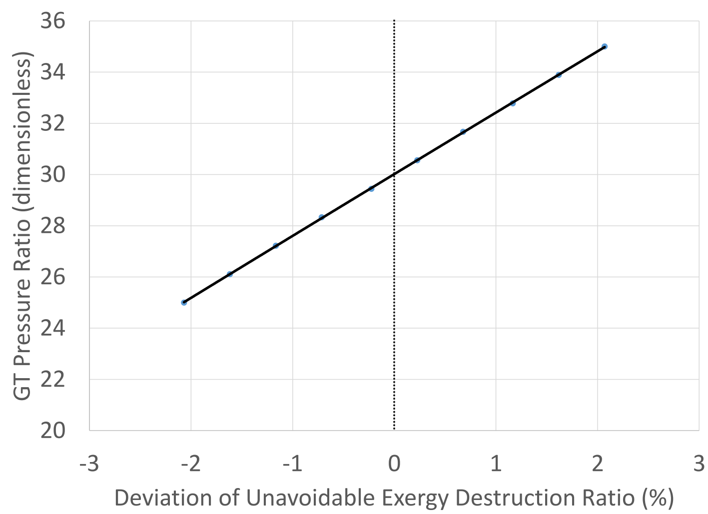

Figure 5.7: Sensitivity of the GT unavoidable exergy destruction ratio to changes in GT pressure ratio

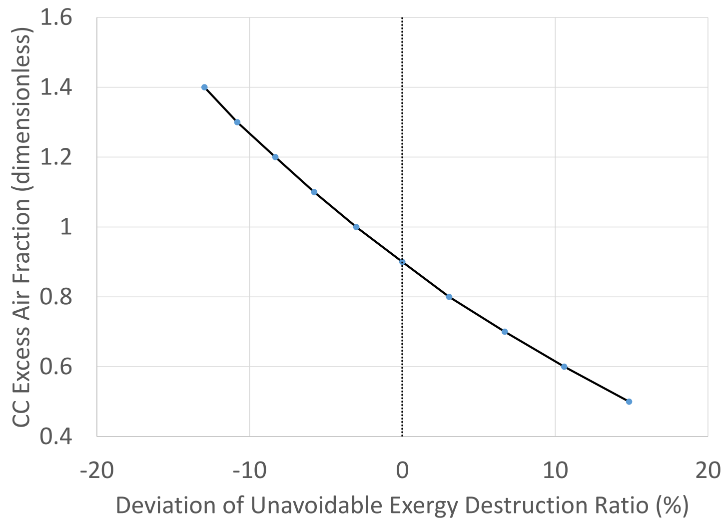

Figure 5.8: Sensitivity of the CC unavoidable exergy destruction ratio to changes in the excess air fraction

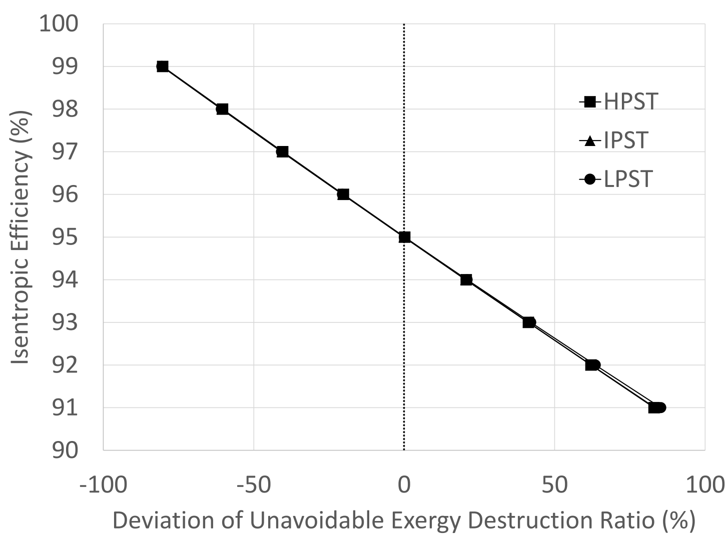

Figure 5.9: Sensitivity of the HPST, IPST, and LPST unavoidable exergy destruction ratio to changes in respective isentropic efficiency

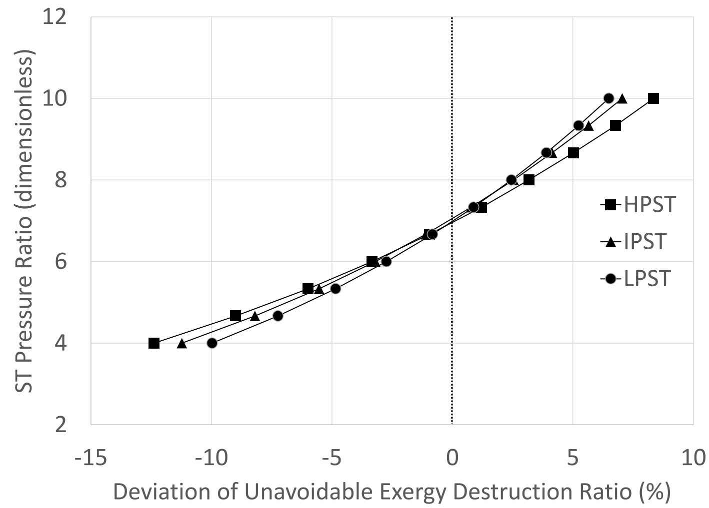

Figure 5.10: Sensitivity of the HPST, IPST, and LPST unavoidable exergy destruction ratio to changes in respective pressure ratio

The results of the sensitivity analysis highlighted the significance of select assumptions. For all of the compressors and expanders, $(\dot{E}_D/\dot{E}_P)_k^{UN}$ was especially sensitive to isentropic efficiency. In the AC (figure 5.4), for example, a mere 1% change in isentropic efficiency resulted in a 20% deviation in $(\dot{E}_D/\dot{E}_P)_k^{UN}$. Illustrated in figure 5.10, all of the ST stages had similar sensitivity to isentropic efficiency, where a 1% change in isentropic efficiency resulted in a 20% deviation in $(\dot{E}_D/\dot{E}_P)_k^{UN}$. Pressure ratio had a less significant impact on the $(\dot{E}_D/\dot{E}_P)_k^{UN}$ for the compressors and expanders. In the GT, for example, a 10% variation in pressure ratio resulted in a less than 2% variation in $(\dot{E}_D/\dot{E}_P)_k^{UN}$. Illustrated in figure 5.8, in the CC, a 10% increase in excess air resulted in approximately a 3% reduction of $(\dot{E}_D/\dot{E}_P)_k^{UN}$.

5.5.2: Sensitivity to the Cost of Fuel

The sensitivity of the results due to the cost of fuel is another important consideration, as the cost of fuel changes almost daily, and the change is sometimes drastic due to global economics. The fuel cost for natural gas was originally \$2.73/MMBtu, and was varied between \$1.73/MMBtu and \$3.73/MMBtu to illustrate its effect on the results of the exergoeconomic analysis. The sensitivity of the fuel cost per unit exergy $c_{F,k}$ of each component to the cost of natural gas is presented in figure 5.11, and the sensitivity of the sum of capital investment and exergy destruction costs $\dot{Z}_k^{CI} + \dot{C}_{D,k}$ of each component to the cost of natural gas is presented in figure 5.12.

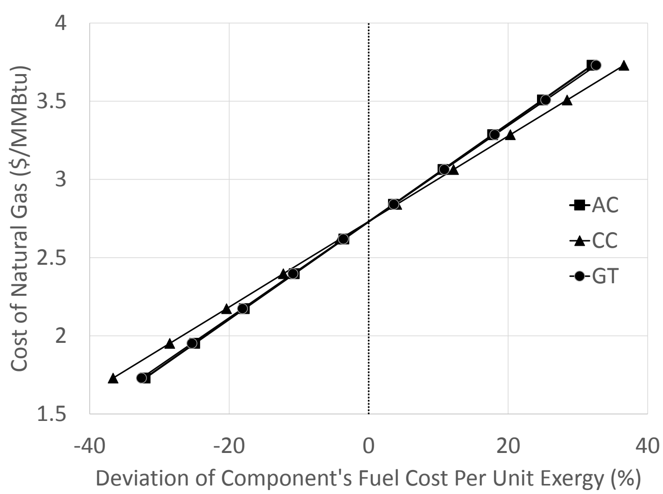

Figure 5.11: Sensitivity of each component’s fuel cost to the cost of natural gas

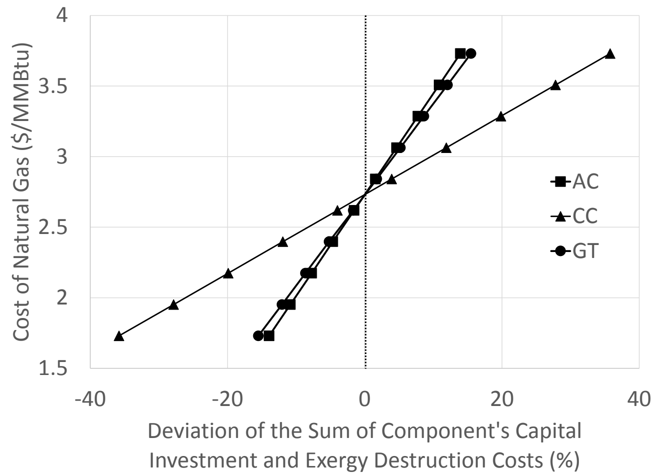

Figure 5.12: Sensitivity of each component's sum of capital investment and exergy destruction costs $\dot{Z}_k^{CI} + \dot{C}_{D,k}$ to the cost of natural gas

Figure 5.11 illustrates that the fuel cost per unit exergy of the AC and GT have similar sensitivity to the cost of natural gas. With a slightly lower slope, it is evident that the CC is slightly more sensitive to the cost of natural gas than the AC and GT. For either component, the results show that the fuel cost per unit exergy is nearly proportional to the cost of natural gas.

From figure 5.12, it is evident that $\dot{Z}_k^{CI} + \dot{C}_{D,k}$ for the AC and GT also have similar sensitivity to the cost of natural gas. With a much lower slope than the AC and GT, it is evident that $\dot{Z}_k^{CI} + \dot{C}_{D,k}$ of the CC is more sensitive to the cost of natural gas than the other components. A 10% variation in the cost of natural gas results in roughly a 10% variation in $\dot{Z}_k^{CI} + \dot{C}_{D,k}$ for the CC. This nearly proportional relationship result for the CC is due to its relatively low capital investment cost, and relatively high cost of exergy destruction.

6. CONCLUSION

The conventional exergy and exergoeconomic methods were well supplemented with the advanced methods, highlighting the significant difference between the exergy destruction and the exergy destruction that can realistically be avoided. The results of the exergetic and exergoeconomic analyses revealed that there is indeed realistic potential for improvement at the CCPP2 power plant. The results of the exergetic analysis identified the combustion chamber as having the largest magnitude of overall exergy destruction, avoidable exergy destruction, and endogenous avoidable exergy destruction. By reducing the air-fuel ratio, and further preheating the combustion air, exergy destruction in the combustion chamber could be reduced by up to 29.25 MW. The results of the conventional exergoeconomic analysis indicated that the combustion chamber was also the most cost-effective component to be improved, due to a low cost of capital investment and high cost of exergy destruction.

From determining exogenous exergy destruction, the significance of component interactions were highlighted. Further analysis could be done by determining how much the exergy destruction in one system component contributes to that of another, and how the interaction of components in the system contributes to the exergy destruction in a single component. The latter is referred to as the mexogenous exergy destruction [2]. With this information, it may be determined that upgrading certain components could significantly reduce the exogenous exergy destruction in other components in the system. Doing so could potentially be more cost-effective than simply improving a component based on its own value of avoidable endogenous exergy destruction.

REFERENCES

[1] "2014 Annual Energy Outlook," EIA. U.S. Energy Information Administration, 16 Dec. 2013. Web. 4 May 2015.

[2] S. Kelly, "Energy Systems Improvement based on Endogenous and Exogenous Exergy Destruction," Ph.D. dissertation, Inst. of Energy Eng., Tech. Univ. of Berlin, Berlin, Germany, 2008.

[3] S. Kelly, G. Tsatsaronis, and T. Morosuk, "Advanced exergetic analysis: Approaches for splitting the exergy destruction into endogenous and exogenous parts," *Energy*, vol. 34, pp. 384–391, 2009.

[4] E. Sciubba and G. Wall, "A brief commented history of exergy from the beginnings to 2004," *Int. J. Thermodyn.*, vol. 10, no. 1, pp. 1–26, 2007.

[5] W. Gibbs, "A Method of Geometrical Representation of the Thermodynamic Properties of Substances By Means of Surfaces," *Transactions of the Connecticut Academy*, vol. 2, pp. 382-404, 1873.

[6] C. Giannantoni, A. Lazzaretto, A. Macor, A. Mirandola, A. Stoppato, S. Tonon, and S. Ulgiati, "Multicriteria approach for the improvement of energy systems design," *Energy*, vol. 30, pp. 1989–2016, 2005.

[7] C. J. Butcher and B. V. Reddy, "Second law analysis of a waste heat recovery based power generation system," *Int. J. Heat Mass Transf.*, vol. 50, pp. 2355–2363, 2007.

[8] C. Ö. Çolpan, "Exergy Analysis of Combined Cycle Cogeneration Systems," M.S. Thesis, Dept of Mech. Eng., Middle East Technical Univ., Ankara, Turkey, 2005.

[9] A. De Sa and S. Al Zubaidy, "Gas turbine performance at varying ambient temperature," *Appl. Therm. Eng.*, vol. 31, no. 14–15, pp. 2735–2739, 2011.

[10] S. C. Kaushik, V. S. Reddy, and S. K. Tyagi, "Energy and exergy analyses of thermal power plants: A review," *Renew. Sustain. Energy Rev.*, vol. 15, no. 4, pp. 1857–1872, 2011.

[11] T. W. Song, J. L. Sohn, J. H. Kim, T. S. Kim, and S. T. Ro, "Exergy-based performance analysis of the heavy-duty gas turbine in part-load operating conditions," *Exergy, An Int. J.*, vol. 2, pp. 105–112, 2002.

[12] A. Acır, A. K. Bilginsoy, and H. Coşkun, “Investigation of varying dead state temperatures on energy and exergy efficiencies in thermal power plant,” *J. Energy Inst.*, vol. 85, no. 1, pp. 14–21, 2012.

[13] I. H. Aljundi, “Energy and exergy analysis of a steam power plant in Jordan,” *Appl. Therm. Eng.*, vol. 29, no. 2–3, pp. 324–328, 2009.

[14] S. C. Gülen and R. W. Smith, “Second Law Efficiency of the Rankine Bottoming Cycle of a Combined Cycle Power Plant,” *J. Eng. Gas Turbines Power*, vol. 132, no. 1, 2010.

[15] S. Can Gülen and J. Joseph, “Combined Cycle Off-Design Performance Estimation: A Second-Law Perspective,” *J. Eng. Gas Turbines Power*, vol. 134, no. 1, 2011.

[16] B. T. Aklilu and S. I. Gilani, “Exergetic Performance Analysis of a Cogeneration Plant at Part Load Operations,” vol. 4, no. 10, 2010.

[17] Y. Haseli, I. Dincer, and G. F. Naterer, “Thermodynamic analysis of a combined gas turbine power system with a solid oxide fuel cell through exergy,” *Thermochim. Acta*, vol. 480, pp. 1–9, 2008.

[18] F. Petrakopoulou, G. Tsatsaronis, T. Morosuk, and C. Paitazoglou, “Environmental evaluation of a power plant using conventional and advanced exergy-based methods,” *Energy*, vol. 45, no. 1, pp. 23–30, 2012.

[19] F. Petrakopoulou, G. Tsatsaronis, T. Morosuk, and A. Carassai, “Conventional and advanced exergetic analyses applied to a combined cycle power plant,” *Energy*, vol. 41, pp. 146-152, 2012.

[20] L. Wang, Y. Yang, T. Morosuk, and G. Tsatsaronis, “Advanced Thermodynamic Analysis and Evaluation of a Supercritical Power Plant,” *Energies*, vol. 5, no. 6, pp. 1850-1863, 2012.

[21] T. Morosuk and G. Tsatsaronis, "Stengths and Limitations of Advanced Exergy Analyses," *IMECE*, vol. 6, 2013.

[22] G. Tsatsaronis, “Combination of Exergetic and Economic Analysis in Energy-Conversion Processes,” *Proceedings, European Congress, Energy Economics and Management in Industry*, vol. 1, pp. 151-157, 1984.

[23] G. Tsatsaronis and M. Winhold, “Exergoeconomic analysis and evaluation of energy-conversion plants—I. A new general methodology,” *Energy*, vol. 10, no. 1, pp. 69–80, 1985.

[24] G. Tsatsaronis and M. Winhold, “Exergoeconomic analysis and evaluation of energy-conversion plants—II. Analysis of a coal-fired steam power plant,” *Energy*, vol. 10, no. 1, pp. 81–94, 1985.

[25] A. Valero, M. A. Lozano, L. Serra, G. Tsatsaronis, J. Pisa, C. Frangopoulos, and M. R. von Spakovsky, “CGAM problem: Definition and conventional solution,” *Energy*, vol. 19, no. 3, pp. 279–286, 1994.

[26] I. Dincer and M. Rosen. “Exergy: Energy, Environment, and Sustainable Development,” Oxford, UK: Elsevier, 2013. Print.

[27] A. Lazzaretto and G. Tsatsaronis, “SPECO: A systematic and general methodology for calculating efficiencies and costs in thermal systems,” *Energy*, vol. 31, pp. 1257–1289, 2006.

[28] F. Cziesla, G. Tsatsaronis, and Z. Gao. “Avoidable thermodynamic inefficiencies and costs in an externally fired combined cycle power plant,” *Energy*, vol. 31, no. 10-11, pp. 1472-1489, 2006.

[29] P. Ahmadi, I. Dincer, and M. Rosen, “Exergy, exergoeconomic and environmental analyses and evolutionary algorithm based multi-objective optimization of combined cycle power plants,” *Energy*, vol. 36, no. 10, pp. 5886–5898, 2011.

[30] R. Karaali, I.T. Oztürk, “Thermoeconomic optimization of gas turbine cogeneration plants,” *Energy*, vol. 80, pp. 474-485, 2015.

[31] “Pearl Street Station.” *Wikipedia: The Free Encyclopedia*, Wikimedia Foundation, Inc., 24 Apr. 2015. Online. May. 2015. <https://en.wikipedia.org/wiki/Pearl_Street_Station>

[32] M. P. Boyce. “Handbook for Cogeneration and Combined Cycle Power Plants,” New York: ASME, 2002. Print.

[33] Davis, D. “Factors which affect Industrial Gas Turbine Air Mass Flow Rate,” Real Time Power, July 1, 2012. Online. Nov, 2014. <http://realtimepowerinc.com/factors-which-affect-gt-air-mass-flow-rate>

[34] Y. A. Çengel, and M. A. Boles. “Thermodynamics: An Engineering Approach,” Boston: McGraw-Hill, 2001. Print.

[35] T. J. Kotas, “The exergy method of thermal power analysis,” London: Butterworths, 1985. Print.

[36] A. Bejan, G. Tsatsaronis, and M. Moran, “Thermal Design and Optimization,” Wiley-IEEE, 1996. Print.

[37] F. Cziesla and G. Tsatsaronis, “Thermoeconomics,” *Encyclopedia of physical science and technology*, Academic Press, 2001.

[38] G. Tsatsaronis, “Definitions and nomenclature in exergy analysis and exergoeconomics,” *Energy*, vol. 32, pp. 249–253, 2007.

[39] T. Morosuk and G. Tsatsaronis, “Advanced exergy analysis for chemically reacting systems-application to a simple open gas-turbine system,” *Int. Journal of* Thermodynamics, vol. 12, no. 3, pp. 105-111, 2009.

[40] G. Tsatsaronis, M. Park, “On avoidable and unavoidable exergy destructions and investment costs in thermal systems,” *Energy Conversion and Management*, vol. 43, no. 9-12, pp. 1259–1270, 2002.

[41] J. Simoes, “Gas Turbine Prices by Output,” NYE Thermodynamics, 2015. Online. 11 Sep 2015. < http://www.nyethermodynamics.com/trader/outprice.htm>

[42] “Reducing Acid Rain,” EPA. U.S. Environmental Protection Agency, 24 Apr 2014. Online, 30 Aug 2015. <http://www.epa.gov/region1/eco/acidrain/reducing.html>

APPENDIX

A-1: Breakdown of capital investment costs

Component	% of total investment (%)	I_k (10^3 US$)
AC	39.40	18218.56
CC	4.30	1988.32
GT	56.31	26037.74

Total investment cost was $46,240,000

A-2: Equations used for calculations in the exergoeconomic analysis

```
"Known Properties"
W_AC=260.6*3.412 [MMBtu/h]
W_GT=463.9*3.412 [MMBtu/h]
E_7=229.22
E_8=734.42
E_9=725.34
E_F=718.9
E_ext=16.96
E_cool=13.94
E_14=197.58
C_1=0
Fuel=2.73 [$/MMBtu]
C_F=Fuel*.021520*109600
E_D_AC=14.45
E_D_CC=(E_7+E_F)-E_8
E_D_GT=33.14
t=8000 [hrs]
beta=.182 "capital recovery factor"
gamma=.001092 "fraction of O&M costs"

"Investment Costs in ($/h) with 39.4% AC, 4.3% CC, 56.3% GT"
CI=200*231.2*1000
I_AC=.394*CI
I_CC=.043*CI
I_GT=.563*CI
Z_AC=(beta+gamma)*I_AC/t
Z_CC=(beta+gamma)*I_CC/t
Z_GT=(beta+gamma)*I_GT/t

"AC Cost Equations"
C_1+c_W*W_AC+Z_AC=C_7+C_ext
C_7/E_7=C_ext/E_ext
```

```
"CC"
C_7+C_F+Z_CC=C_8

"TCS"
C_8=c8*E_8
C_9=c9*E_9
c8=c9

"GT"
C_9+C_ext+Z_GT=c_W*W_GT+C_14
C_14=E_14*(C_9+C_ext)/(E_9+E_ext)

"Fuel and Product Costs"
c_F_AC=c_W+C_14/E_14
c_P_AC=(C_7+C_ext)/(E_7+E_ext)
C_D_AC=c_F_AC*E_D_AC

c_F_CC=C_F/E_F
c_P_CC=(C_8-C_7)/(E_8-E_7)
C_D_CC=c_F_CC*E_D_CC

c_F_GT=(C_9+C_ext+C_14)/(E_9+E_cool+E_14)
c_P_GT=c_W+C_14/E_14
C_D_GT=c_F_GT*E_D_GT

sum_AC=Z_AC+C_D_AC
sum_CC=Z_CC+C_D_CC
sum_GT=Z_GT+C_D_GT

"Splitting the Cost Rates"
"Endogenous/Exogenous/Avoidable/Unavoidable"
E_D_AC_EN=8.85
E_D_AC_EX=5.6
E_D_AC_AV=9.1
E_D_AC_UN=5.35
E_D_AC_EN_AV=6.27
E_D_AC_EN_UN=2.58
E_D_AC_EX_AV=2.84
E_D_AC_EX_UN=2.76
E_D_CC_EN=130
E_D_CC_EX=84
E_D_CC_AV=45.46
E_D_CC_UN=168.54
E_D_CC_EN_AV=29.25
E_D_CC_EN_UN=100.75
E_D_CC_EX_AV=16.21
E_D_CC_EX_UN=67.79
E_D_GT_EN=19.43
E_D_GT_EX=13.71
E_D_GT_AV=15.56
E_D_GT_UN=17.58
E_D_GT_EN_AV=7.8
```

```
E_D_GT_EN_UN=11.63
E_D_GT_EX_AV=7.74
E_D_GT_EX_UN=5.97

C_D_AC_EN=c_F_AC*E_D_AC_EN
C_D_AC_EX=c_F_AC*E_D_AC_EX
C_D_AC_UN=c_F_AC*E_D_AC_UN
C_D_AC_AV=c_F_AC*E_D_AC_AV
C_D_AC_EN_AV=c_F_AC*E_D_AC_EN_AV
C_D_AC_EN_UN=c_F_AC*E_D_AC_EN_UN
C_D_AC_EX_AV=c_F_AC*E_D_AC_EX_AV
C_D_AC_EX_UN=c_F_AC*E_D_AC_EX_UN

C_D_CC_EN=c_F_CC*E_D_CC_EN
C_D_CC_EX=c_F_CC*E_D_CC_EX
C_D_CC_UN=c_F_CC*E_D_CC_UN
C_D_CC_AV=c_F_CC*E_D_CC_AV
C_D_CC_EN_AV=c_F_CC*E_D_CC_EN_AV
C_D_CC_EN_UN=c_F_CC*E_D_CC_EN_UN
C_D_CC_EX_AV=c_F_CC*E_D_CC_EX_AV
C_D_CC_EX_UN=c_F_CC*E_D_CC_EX_UN

C_D_GT_EN=c_F_GT*E_D_GT_EN
C_D_GT_EX=c_F_GT*E_D_GT_EX
C_D_GT_UN=c_F_GT*E_D_GT_UN
C_D_GT_AV=c_F_GT*E_D_GT_AV
C_D_GT_EN_AV=c_F_GT*E_D_GT_EN_AV
C_D_GT_EN_UN=c_F_GT*E_D_GT_EN_UN
C_D_GT_EX_AV=c_F_GT*E_D_GT_EX_AV
C_D_GT_EX_UN=c_F_GT*E_D_GT_EX_UN
```